

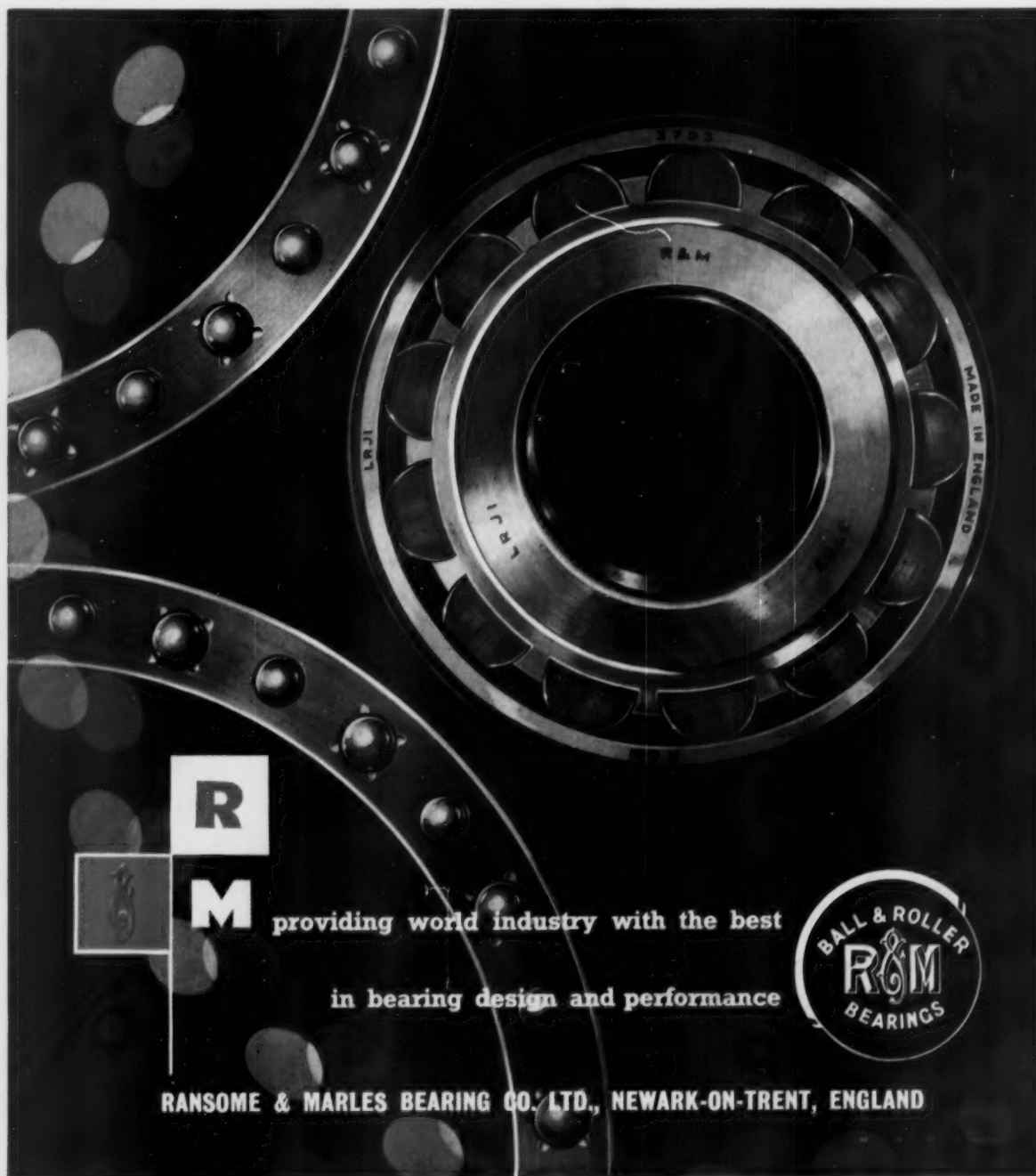
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 46 No. 5

MAY 1956

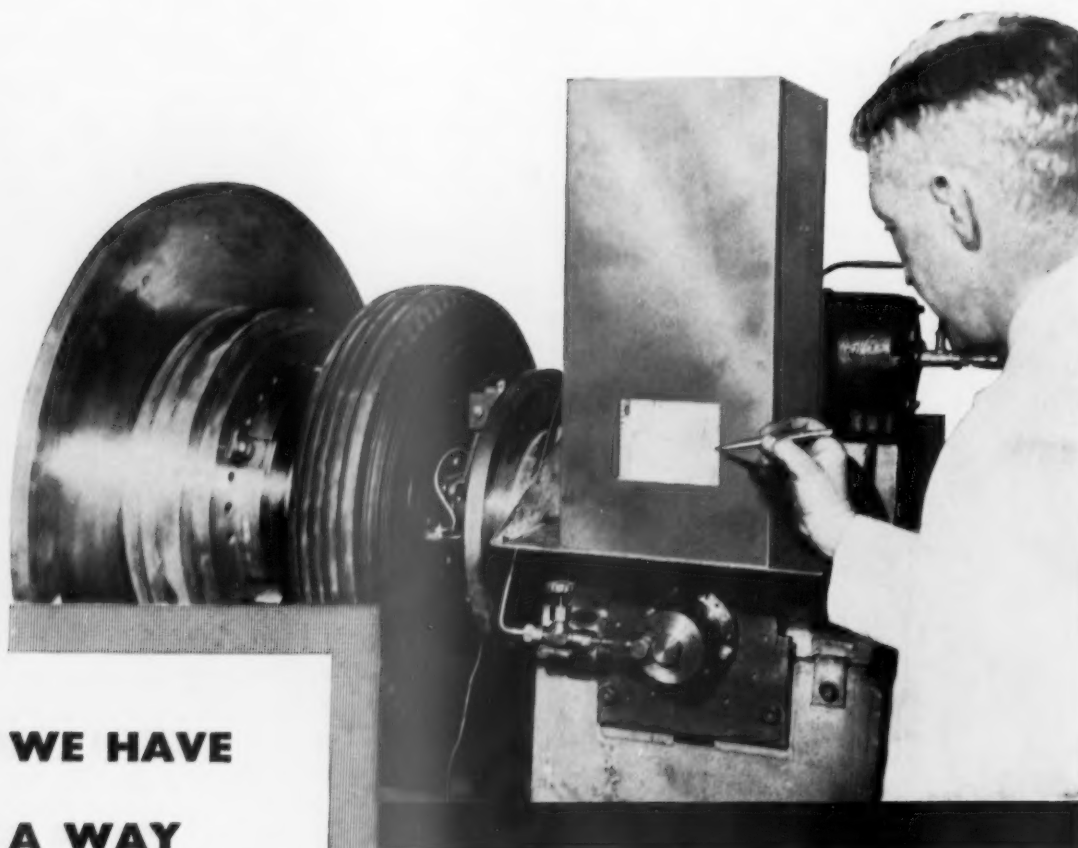
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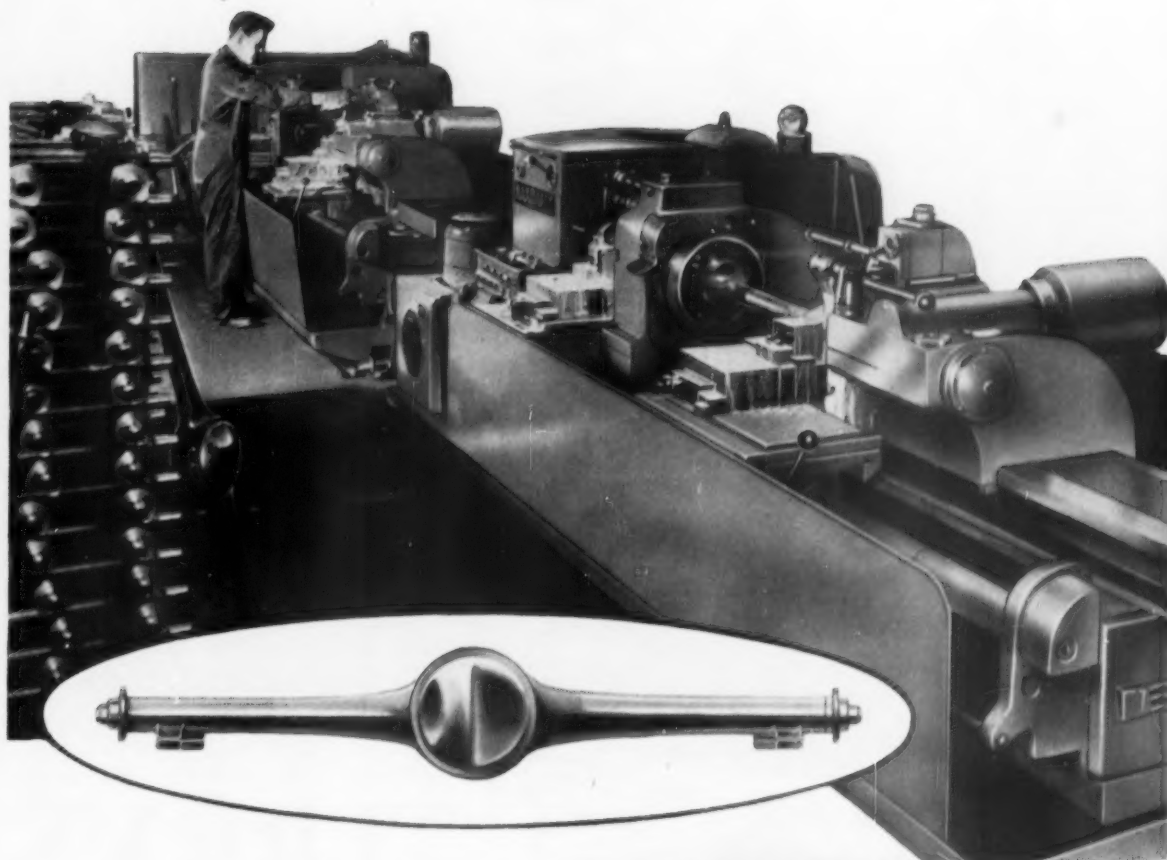
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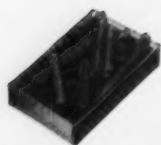
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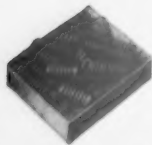
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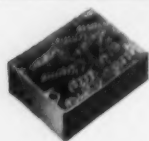
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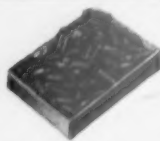
No. 760. 3 doz. Assorted Light Compression Springs 1" to 4" long, 22 to 18 S.W.G., $\frac{1}{8}$ " to $\frac{1}{4}$ " diam. 6/6 each.



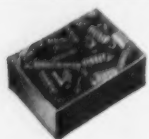
No. 98A. 3 doz. Assorted 1" to 4" long, $\frac{1}{8}$ " to $\frac{1}{4}$ " diam., 19G to 15G. 5/6 each.



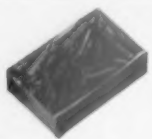
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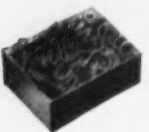
No. 388. $\frac{1}{4}$ gross Assorted Small Expansion Springs. $\frac{1}{8}$ " to $1\frac{1}{2}$ ", 18G to 21G. 9/6 each.



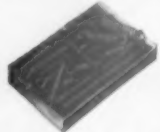
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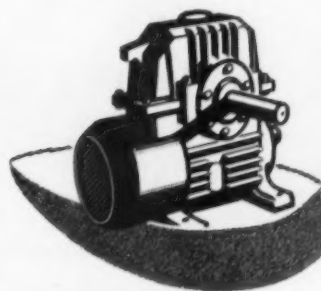
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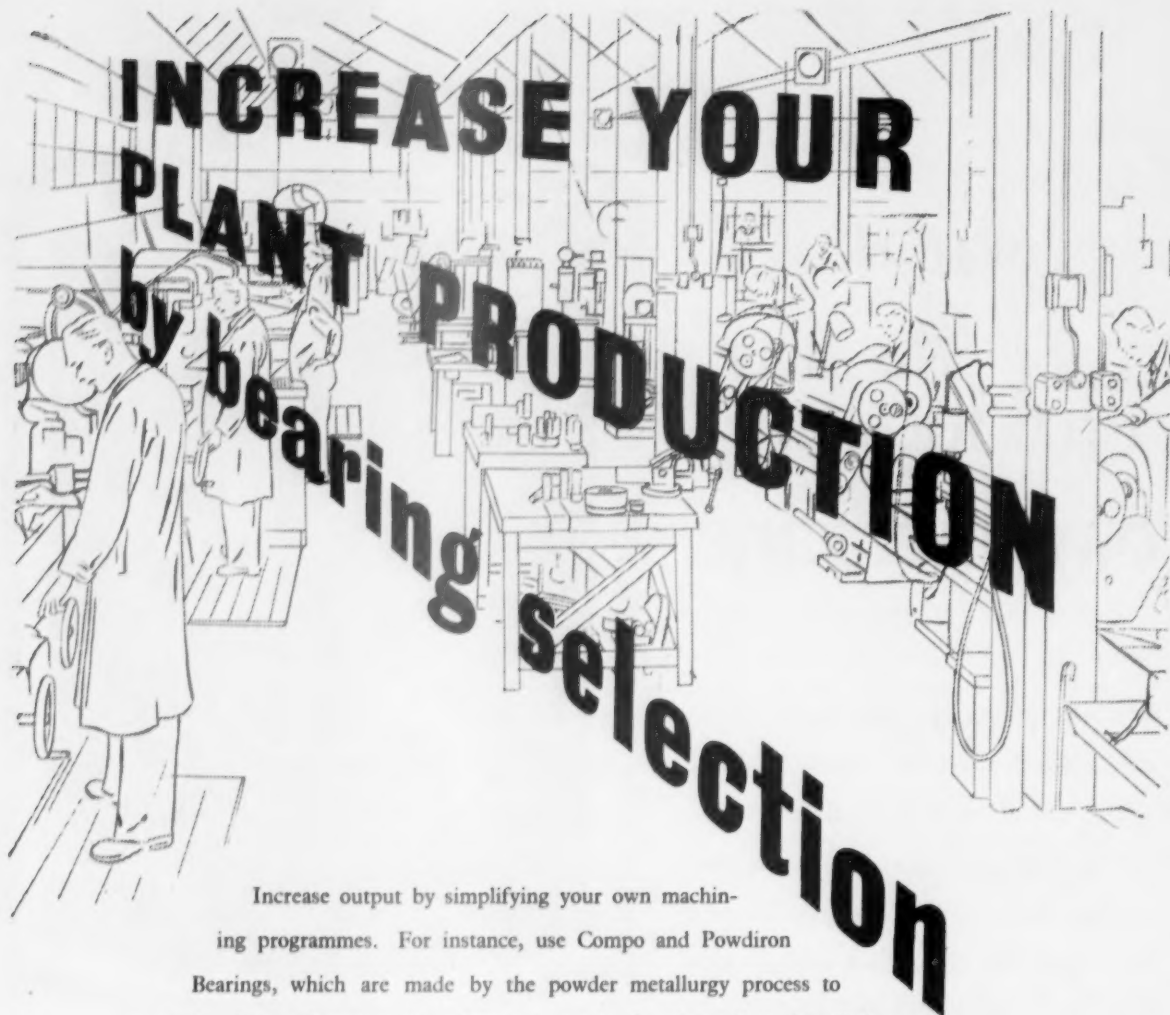
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Automobile Engineer, May 1956

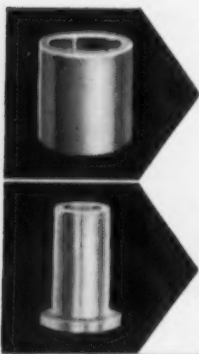
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Increase output by simplifying your own machining programmes. For instance, use Compo and Powdiron Bearings, which are made by the powder metallurgy process to such close tolerances that they are ready for immediate installation.

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LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

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The vacuum-operated servo-unit, having a vacuum valve hydraulically operated directly the brake pedal is pressed, causes the servo to boost the pressure to the wheel cylinders, but always proportionately to the pedal effort.

Should the engine stall or the vacuum fail, the pedal-operated normal hydraulic system remains operative.

Fully patented



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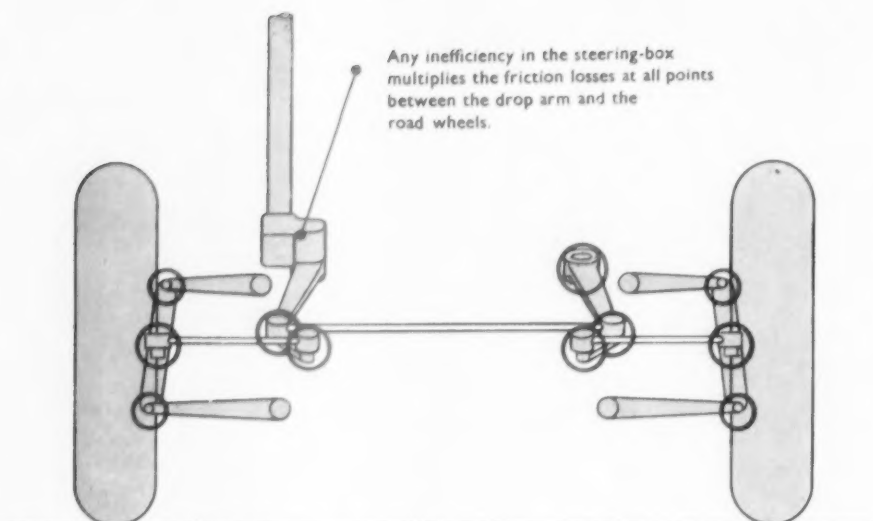
Lockheed

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AUTOMOTIVE PRODUCTS COMPANY LIMITED, LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

MAKING STEERING LIGHTER

Steering-Box Efficiency



The fact that the steering-box is the last link at the driver's end of the chain means that it exercises its inefficiency on all the other losses as well. This makes its efficiency very important.

A theory which has been put forward frequently and is probably still widely held is that if a steering-box is too efficient then joggle of the steering wheel due to road fight will be too pronounced. This probably dates from the days of front axles, when there was little that could be done about the gyroscopic torques that produce joggle (which were accentuated by progressive increases in rotating inertia, as bigger and better balloon tyres became popular) and the only hope of preventing the driver feeling the joggle was to bottle it up. Today this position no longer exists; a large majority of cars are fitted with the double wishbone independent front suspension which allows the amount of gyroscopic torque to be controlled by the layout. Joggle can therefore be reduced at its source and that excuse for an inefficient steering-box can no longer be given.

We may in passing mention that the process of bottling up had two parts; the provision of flexibility in the steering linkage, which reduced the transmission of joggle to the steering-box, and the provision of means to reduce as far as possible the reverse efficiency of the box. The first, the provision of steering linkage flexibility, merits fuller consideration elsewhere; the second has been a dream of many people, including the great Dr. F. W. Lanchester. As we

have said, it is today no longer necessary, and this fact is one of many which allow better steering to be offered today than before.

To substantiate this statement we must first agree what constitutes good steering. This can be logically defined as that which causes the driver as little fatigue as possible. In this connection the reduction of physical fatigue by reducing effort is an obvious improvement. It is at least equally important, however, to help the car to steer itself; complete irreversibility of a steering would mean the complete absence of road feel, and the complete absence also of self-centring or straightening out after a corner. This is tiring both mentally, because of the incessant vigilance demanded of the driver, and physically because of the continued efforts demanded to put into effect the corrections shown to be necessary by that vigilance. Friction in the steering column is one way of reducing the reverse efficiency of a steering-box and it does, by observation, tend to produce the effects described above.

There has been a fairly general realization, with the passage of time, that rolling action rather than rubbing will give higher mechanical efficiencies in steering-boxes and if we look at the types of box used today we shall be able to observe the results of that realization. Let us hope that even higher efficiencies, and still better steering will appear and continue to be with us.

Thompson
Self-adjusting

STEERING AND SUSPENSION JOINTS

AUTOMOTIVE PRODUCTS COMPANY LIMITED, LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

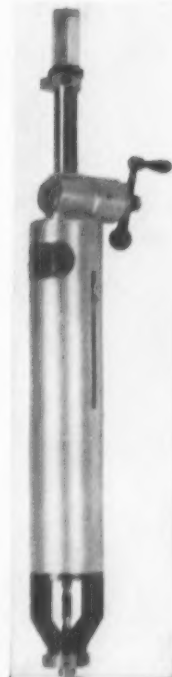
*"And Sisyphus
an huge round stone
did reel against
an hill . . ."*



Poor old Sissy—condemned for ever to a job that got nowhere. Just a legend? Ah, but pause a moment, dear friends. Are there not Sissies in many a works trying this very day to bore half-inch holes through tough metal with the wrong tool?

Alas, alack, there are indeed. Indeedy yes!

The right tool? I am glad you asked me that! It is the Desoutter Pneumatic Rackfeed Drill which you *clamp* into position and operate simply by twiddling the small handle between the thumb and the forefinger. Send at once for exciting art photographs (Genuine students only).



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Wickman Automatics.
Wickman Optical Profile Grinding Machines.
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Pines Tube Benders.
Tibo-Yoder Tube Mills and Roll Forming Equipment.
Weingarten Presses.
Weipert Lathes.
Wickman-Moulton Thread Milling Machines.

STAND 119

National Hall

Davenport Multi-spindle Automatics.
Warner & Swasey Chucking Lathes.

STAND 110

National Hall

Losenhausenwerk Balancing and Testing Equipment.

STAND 318

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Perliff automatic vapour degreasing is absolutely safe; there is no danger to the operator.

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The Perliff is available in 3 sizes.

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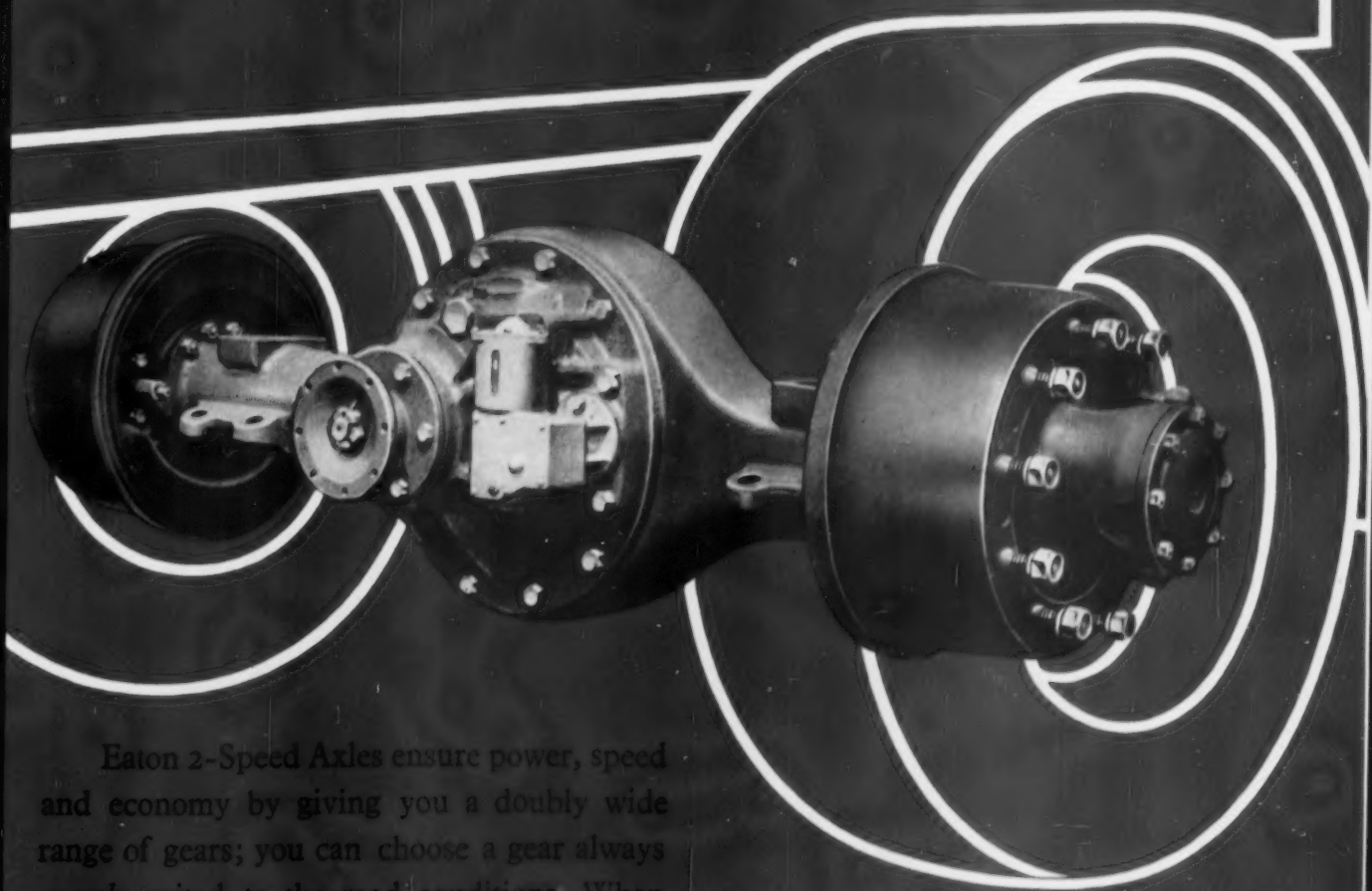
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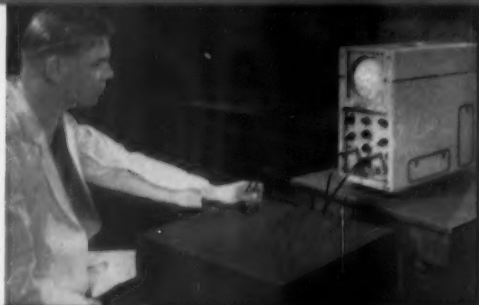


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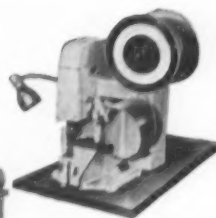
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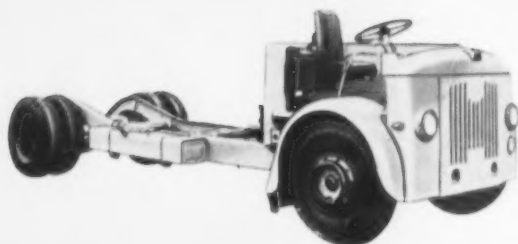
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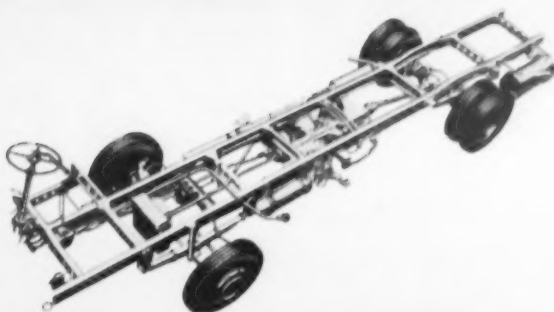
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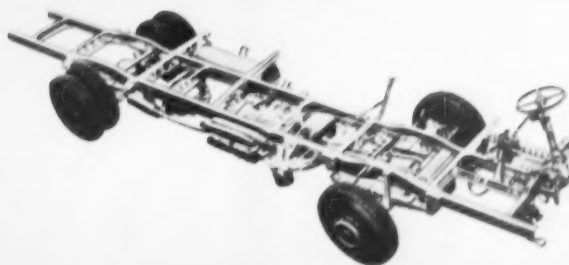
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"Newallastic" bolts and studs have qualities which are absolutely unique. They have been tested by every known device, and have been proved to be stronger and more resistant to fatigue than bolts or studs made by the usual method.

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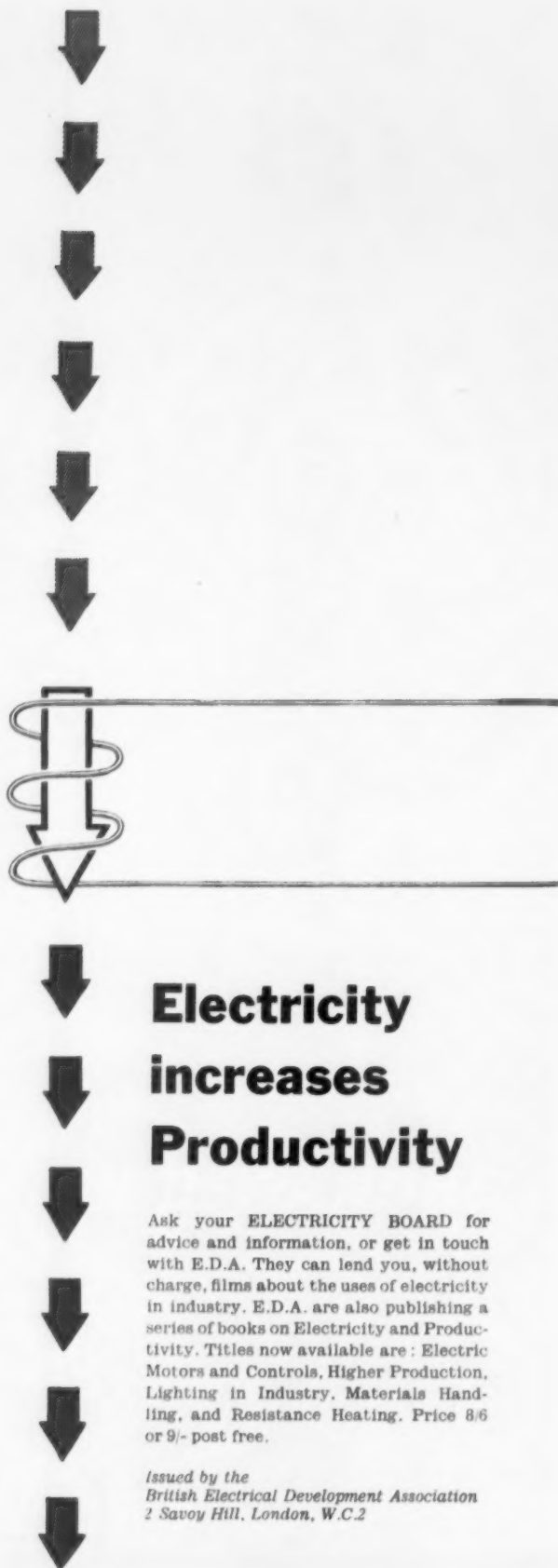
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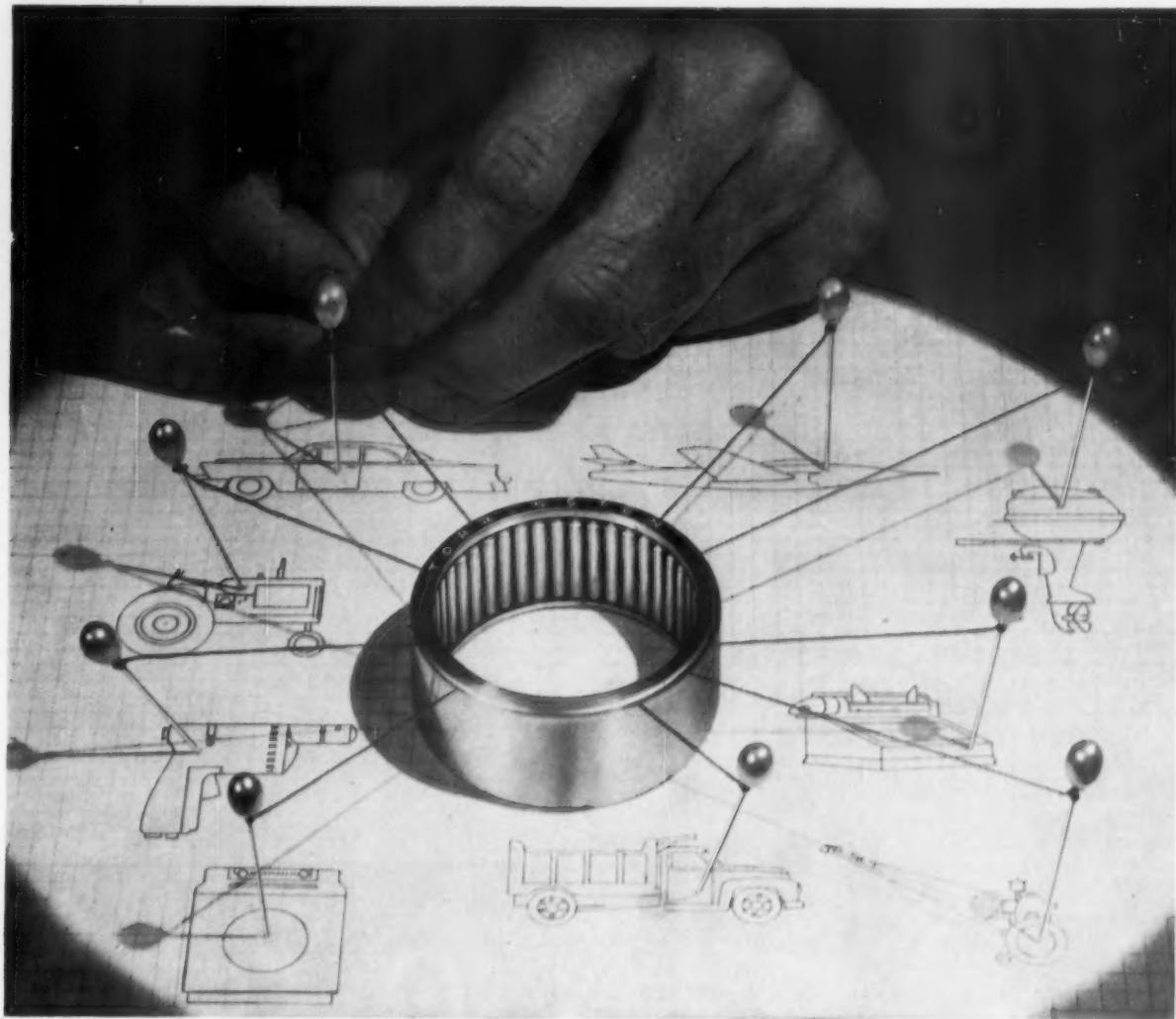
Heat where you want it, perfect control, cleanliness, lower labour costs and better working conditions are some of the advantages of using electricity for heating. *Heating is just one of the many ways in which electricity plays a vital part in the drive for greater productivity.*



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Wherever the Torrington Needle Bearing is put to work, its unique combination of high radial load and small size enables designers and manufacturers to make their products stronger, lighter and more compact. That's why this versatile bearing has become "standard equipment" in thousands of products throughout industry.

In the twenty years that the Needle Bearing has been made, Torrington's Engineering Department has accumulated many lifetimes of experience in adapting the bearing's advantages to products as diverse in size and use as adding machines and tractors. We are always glad to make this experience available to you. *It can be obtained nowhere else.* We invite you to put your bearing problems to our Bearings Division.

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THESE features make the TORRINGTON NEEDLE BEARING unique

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- ★ Compactness and light weight
- ★ Runs directly on hardened shafts
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ΦB



Photograph by courtesy of 'Motor Racing'

The fastest T.T. of all time!
Stirling Moss comes home first in the R.A.C. International T.T. at Dundrod — with an average speed of 88.32 m.p.h. Well done, Stirling Moss and Mercedes!

Brilliant driving by Wharton.
With BP Super in the tank of his Ford Zephyr, Ken Wharton was placed first in the 2001 to 3000 c.c. class in the Production Touring Car Race at the Daily Express Trophy meeting at Silverstone.

A YEAR OF VICTORIES FOR BP SUPER

From Silverstone to Sweden, this outstanding petrol had a remarkable winning record in the year's big races

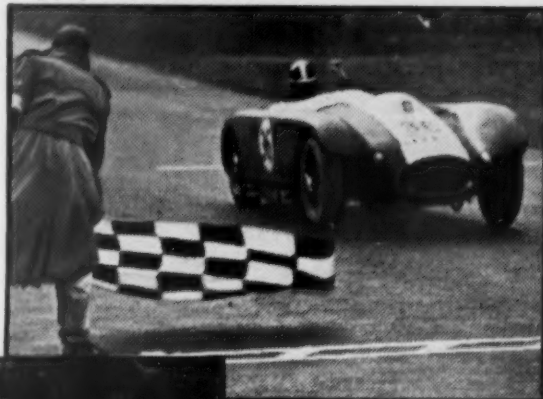
At Silverstone, at Dundrod in Ulster, at Oulton Park and Brands Hatch, at Kristianstad in Southern Sweden, the leading drivers of 1955 battled for victory in some of the world's most exacting races — and the cars that won were fuelled on BP Super.

These successes show what superb performance you can get from a super petrol. Whatever type of car you run, and whatever kind of motoring you go in for, from races and rallies to everyday driving in town and countryside, BP Super gives your engine more 'zip', smoother acceleration and greater freedom from engine-knock. What is more, it gives you more miles per shilling, too.

Big wins in last year's big races have proved the worth of BP Super under the most testing conditions a motorist could encounter. You can see what they mean when they call BP Super 'the petrol with more energy per gallon'!



Photograph by courtesy of 'The Autocar'



Photograph by courtesy of 'The Autocar'

Some of the big wins on BP Super during the 1955 season

Ulster T.T. 1st. Stirling Moss. 2nd and 3rd. J. M. Fangio and G. von Tripps. All driving Mercedes cars.

R.A.C. Rally of Great Britain. 1st. J. Ray and B. Horrocks, Standard. 3rd. K. Richardson and J. Heathcote, Standard. Team award — Standard team and three class wins.

Silverstone Production Touring Car Race. 2001-3000 c.c. class. 1st. Ken Wharton in a Ford Zephyr.

Tulip Rally. 1st. W. J. J. Tak in a Mercedes.

British Empire Trophy. 1st. W. A. Scott-Brown in a Lister-Bristol.

Swedish Grand Prix. 1st. J. M. Fangio. 2nd. Stirling Moss. Both driving Mercedes cars.

Lyons-Charbonnières Rally. Outright Winner — Houel, in an Alfa-Romeo. First four places in general classification. Five firsts out of six other class events.



Here comes another BP Superman! W. A. Scott-Brown used BP Super in his two-litre Lister-Bristol and won the British Empire Trophy for sports cars — besides many other races during the season.

First four places on BP Super. In the important Lyons-Charbonnières Rally the first four places in the general classification were won on BP Super. So were five firsts out of six other class events. Here is Houel, the outright winner, in his Alfa-Romeo — in which he used both BP Super petrol and BP Energol motor oil.



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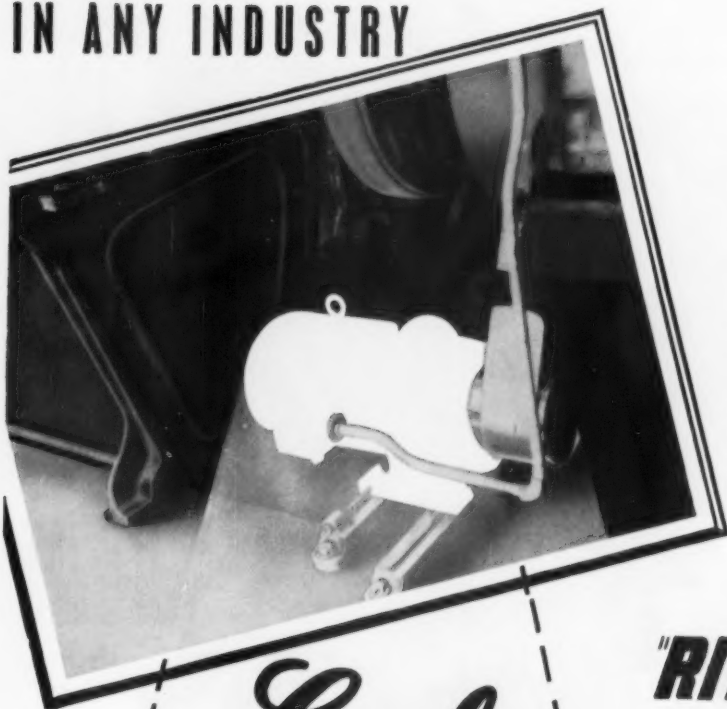
SECOND: the swarf may have been produced by high speed steel tools containing Molybdenum, which is being used to an increasing extent in this country, (in the United States 80-90% contain Molybdenum). On both counts you can rely on Molybdenum, which for more than 10 years has been free from violent price fluctuation.

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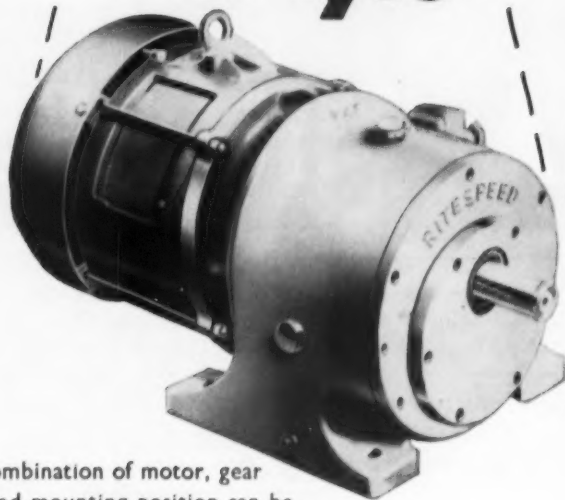


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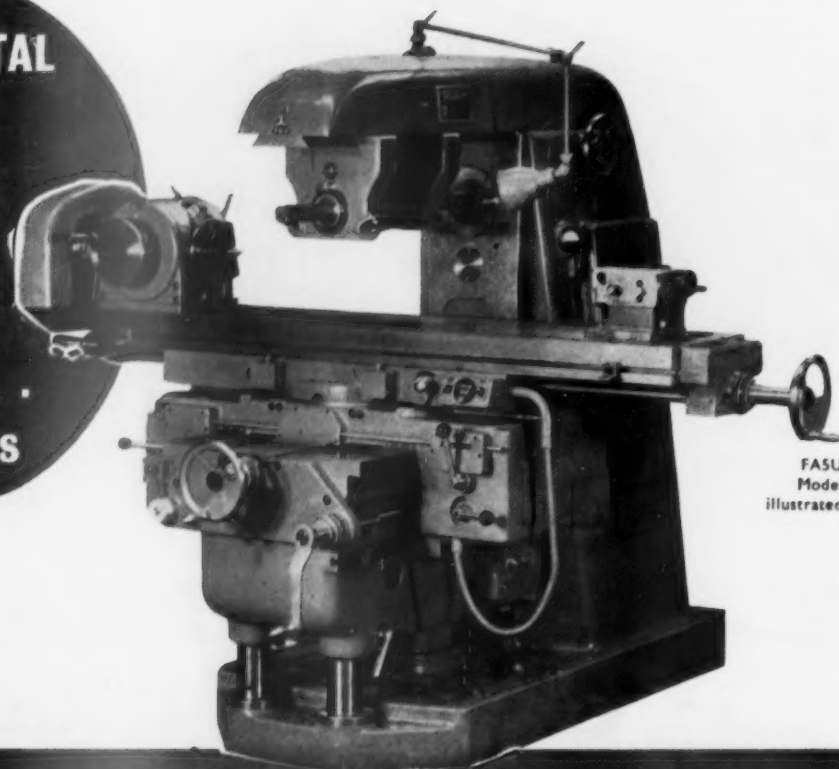
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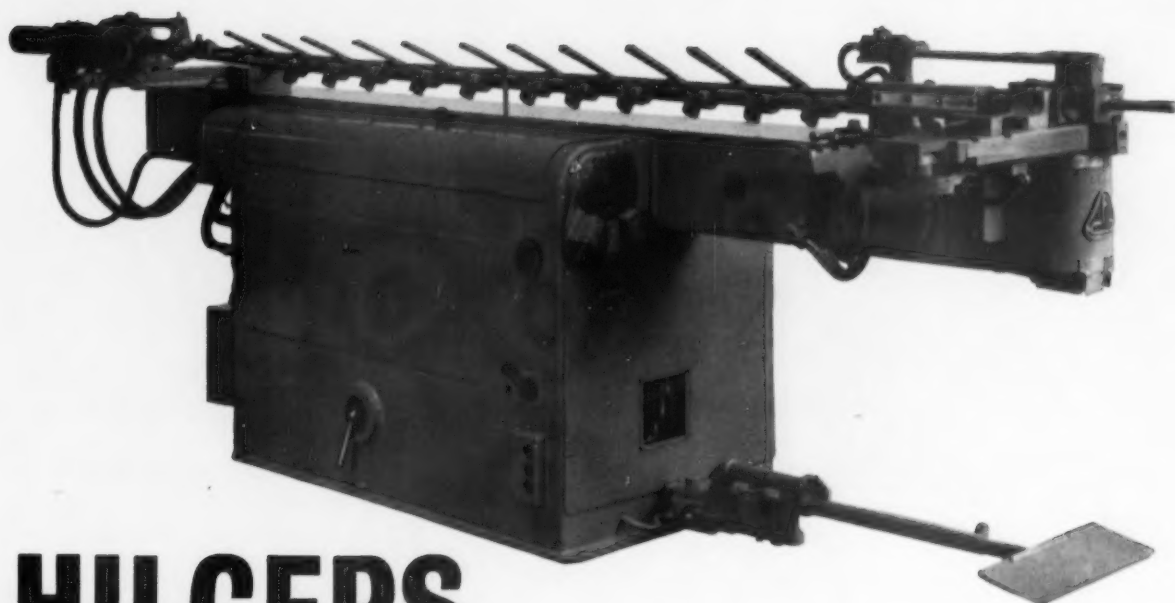
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Power cross travel (approx.)	8"	9"	12"	16"
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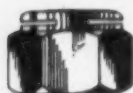
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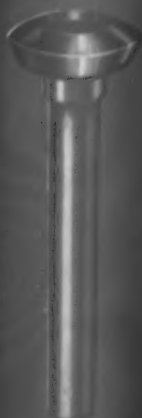
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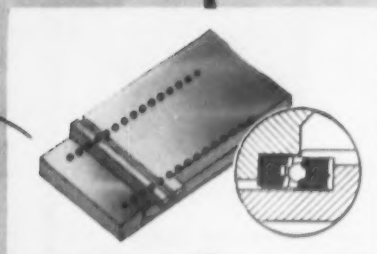
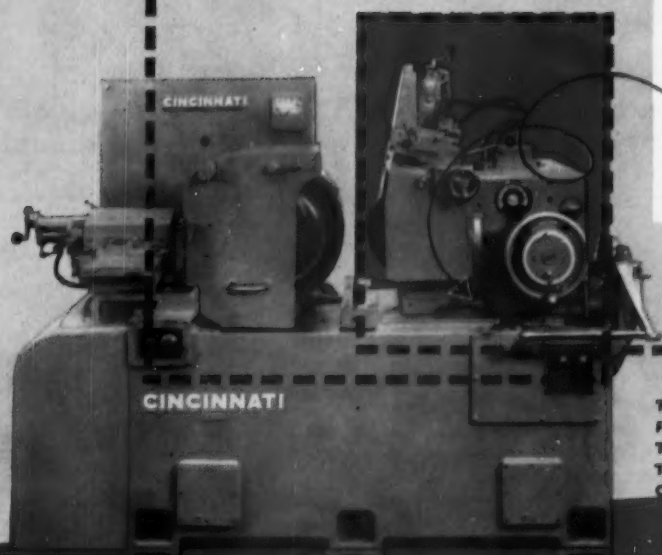
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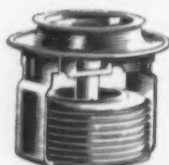


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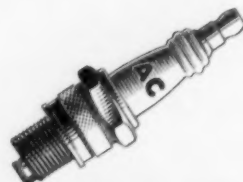
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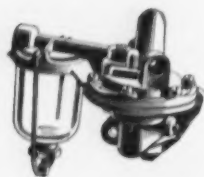
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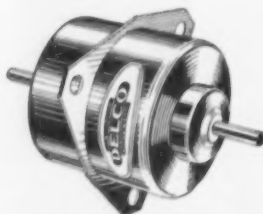
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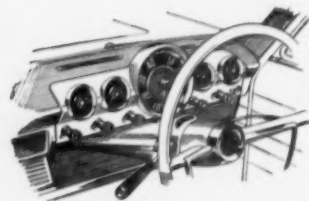
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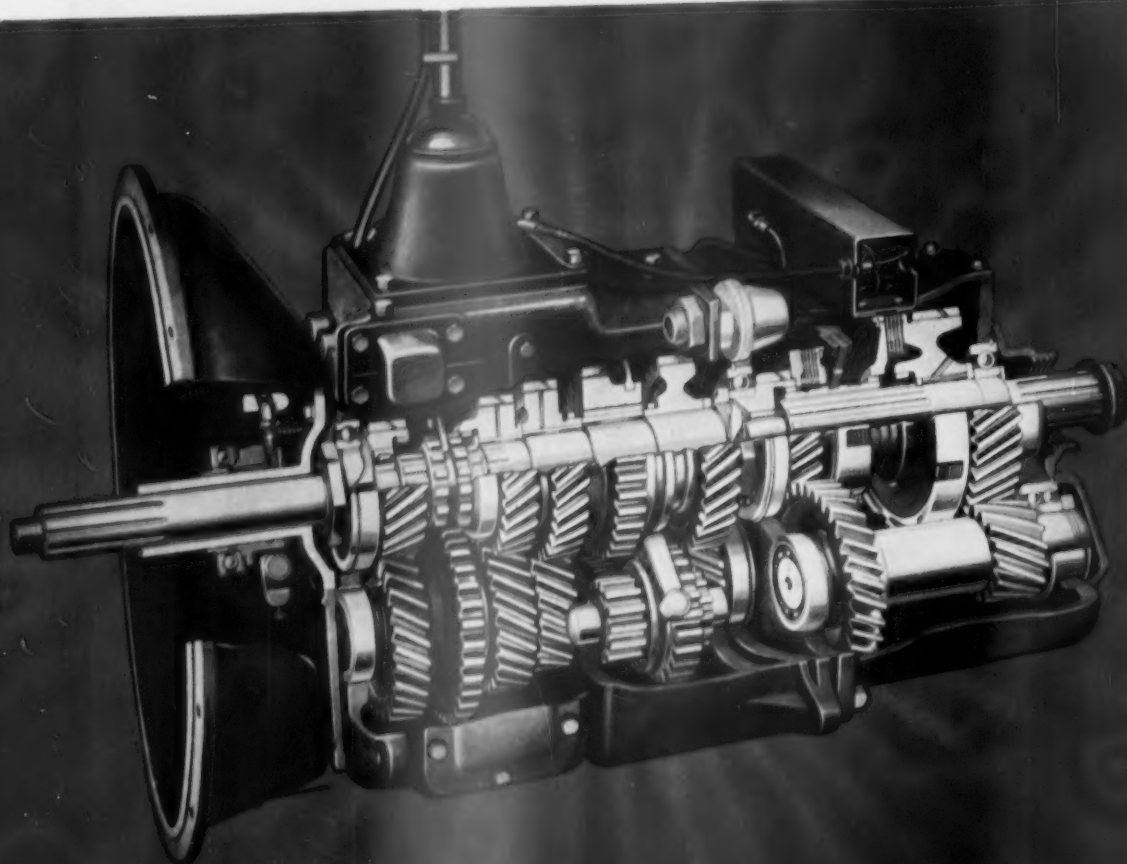
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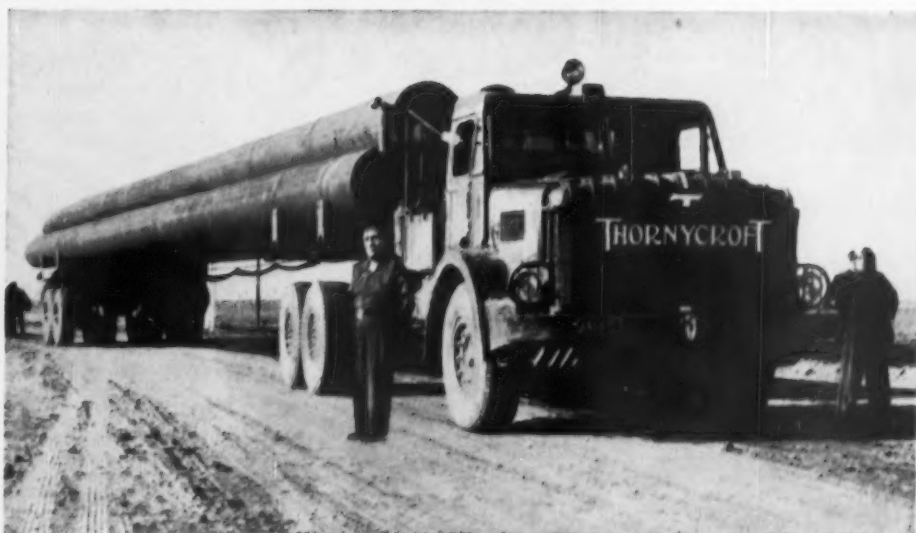


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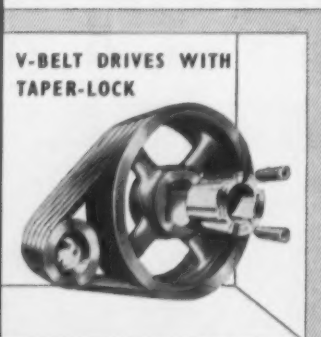
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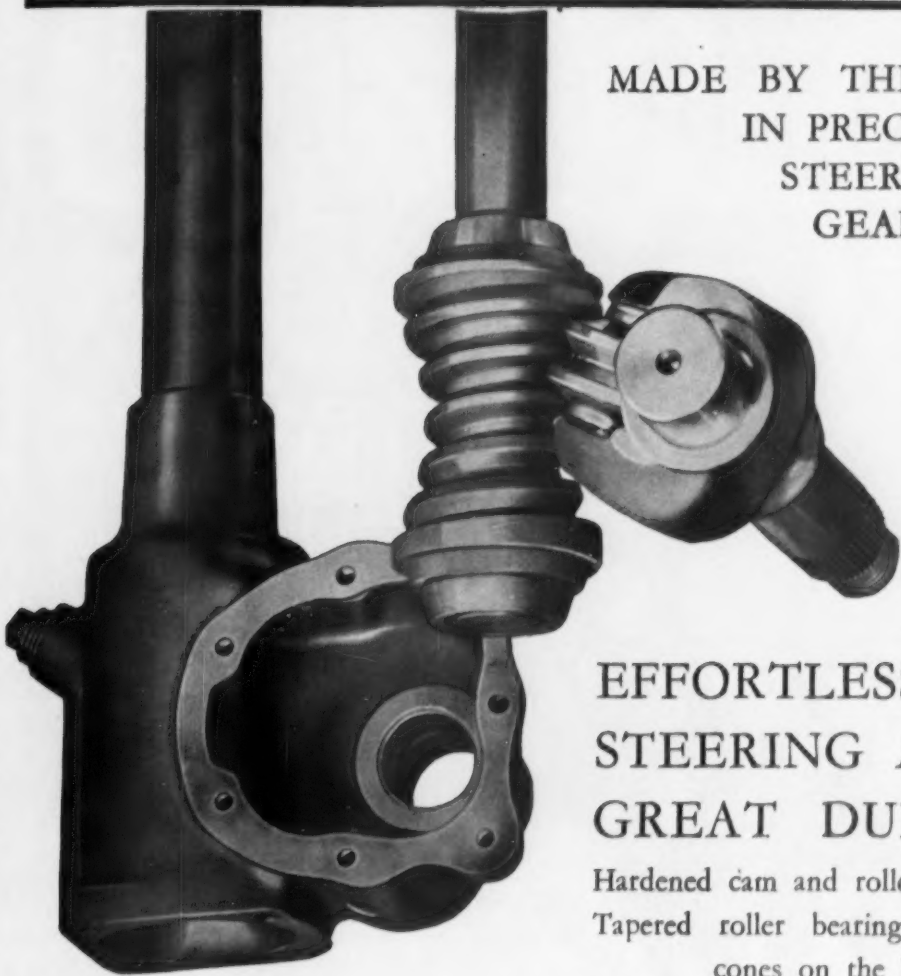
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TIS THE ROUNDHEADS, Sir Percy. Haste thee, Milord — jump for it !”

“But 'tis dark outside,” said Sir Percy plaintively — off the battlefield he was a man of cautious disposition — “'tis wondering where to jump I am ?”

“'Tis no time to ponder” retorted the Squire, “Haste !”

“Very well then, I suppose thou knowest best.” Sir Percy grasped his sword, swung his cloak, and leapt into a world of darkness ; he was pleasantly surprised to land on something soft and yielding. “Gad !” he beamed, “the age of miracles is not yet past.” From beneath him, oddly muffled, the Squire's voice choked ; “'Twould be uncommon kind of milord to remove his scabbard from the small of my back.”

“Faith !” cried the cavalier — “here is a waggish situation indeed. Meseems I made a splendid landing on thy broad and faithful back. And where didst thou come to earth, old friend ?”

The Squire squirmed uneasily. “Varily I could not swear to it, but if I mistake me not, we are in the cattleyard.” His movements were accompanied by an odd squelching sound. “Varily thou wert unfortunate,”

said Sir Percy with soldierly sympathy — “perhaps that will teach thee the folly of a spring in the dark. And speaking of springs” — he spake with a knowledge surprising in one of that period — “Didst know that the most inferior springs may look exactly like a genuine high precision job, guaranteed for years of high performance ? What ! not interested in springs ? Ah, well, perhaps a tub would suit thee better.” He ambled off into the darkness . . .

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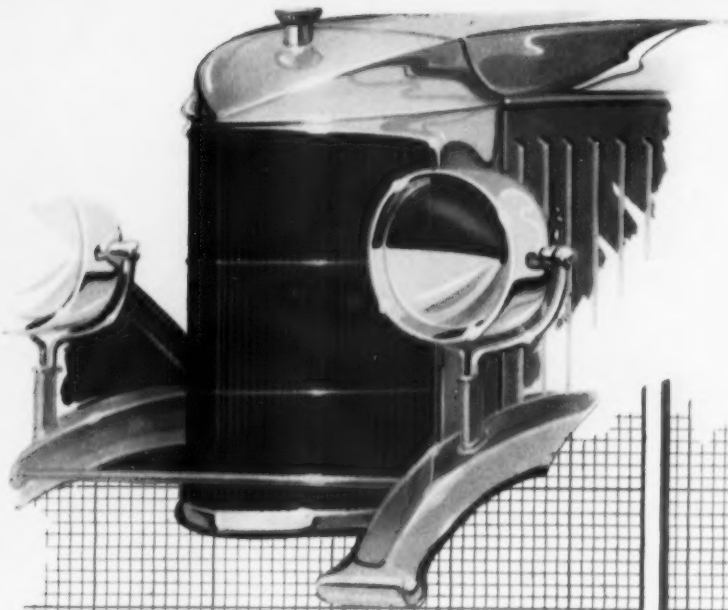
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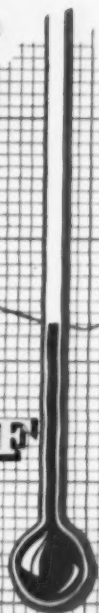
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430C	90	4
437	133	4
* 542	205	5
* 45	250	5
* 045	250	5
† S450	300	4
* 557CM	350	5

2-speed epicyclic unit—either overdrive or underdrive, suitable for 350 lb. ft. maximum engine torque.

* These units are available with either an overdrive or direct top gear.

† Synchromesh for all speeds.

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DAVID BROWN
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TELAFLO DAMPERS

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Sapphire
346



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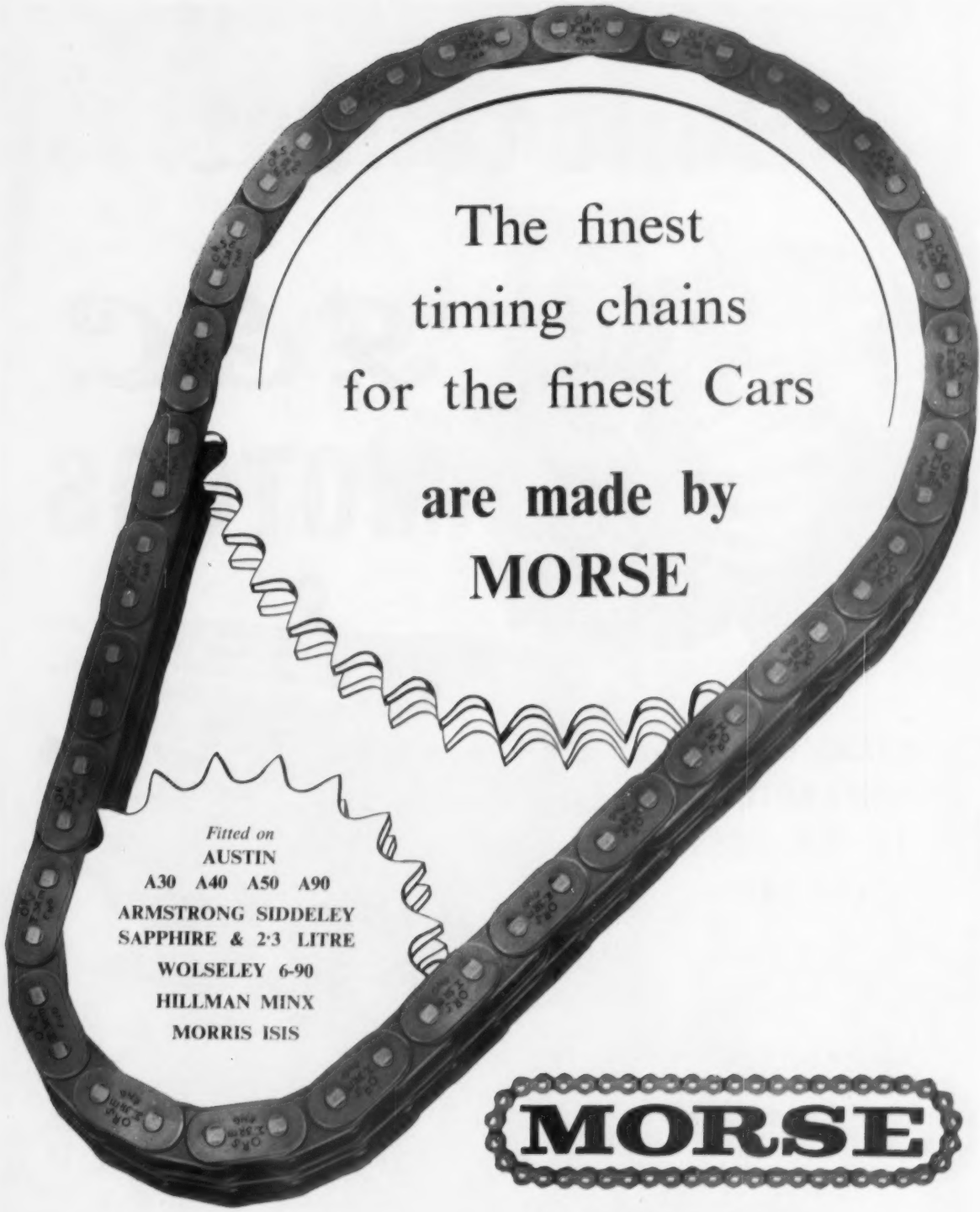
Vide "The Field," February 2, 1956

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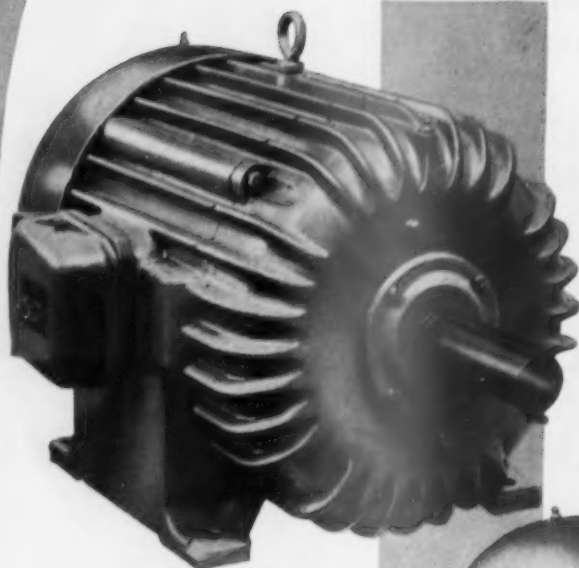
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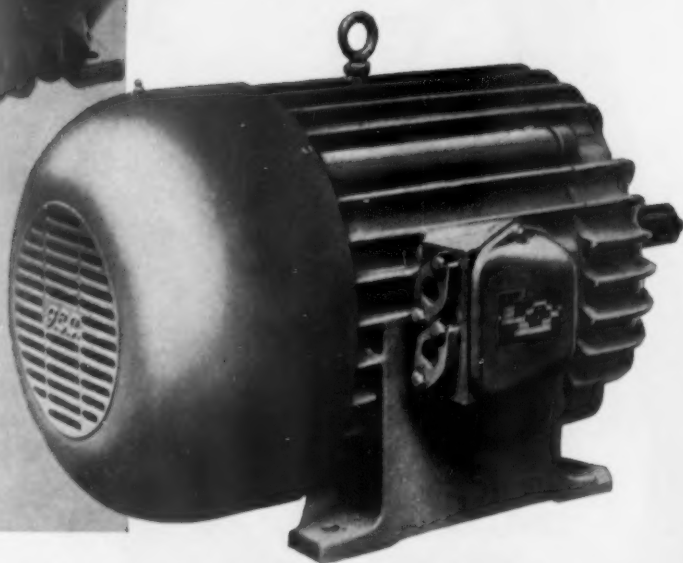


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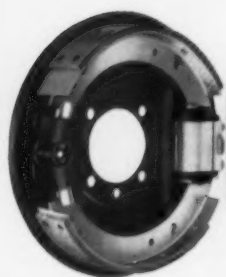
the short
cut is

"SPEEDICUT"

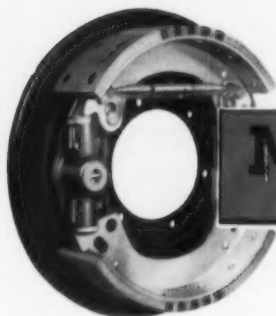
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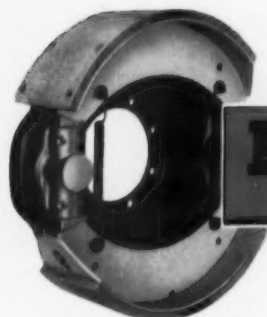
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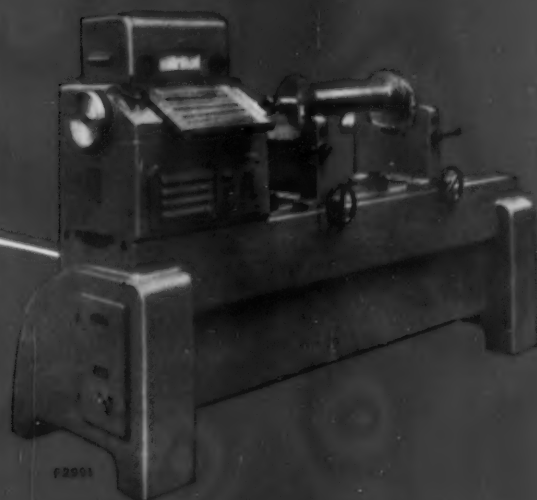
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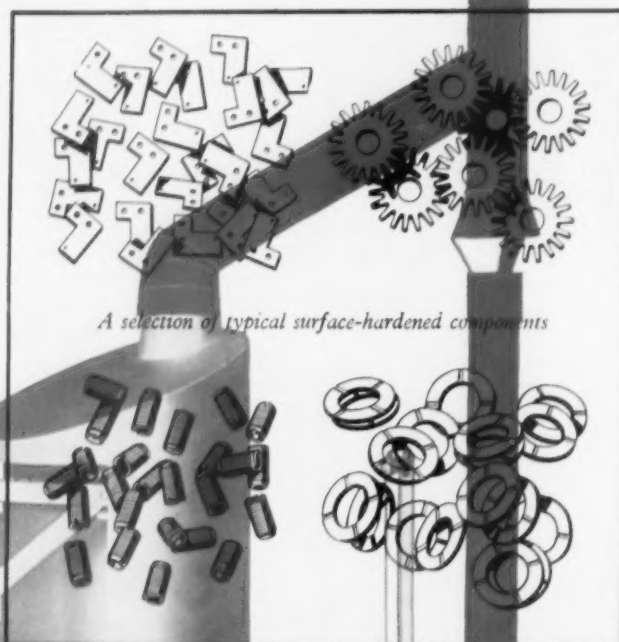
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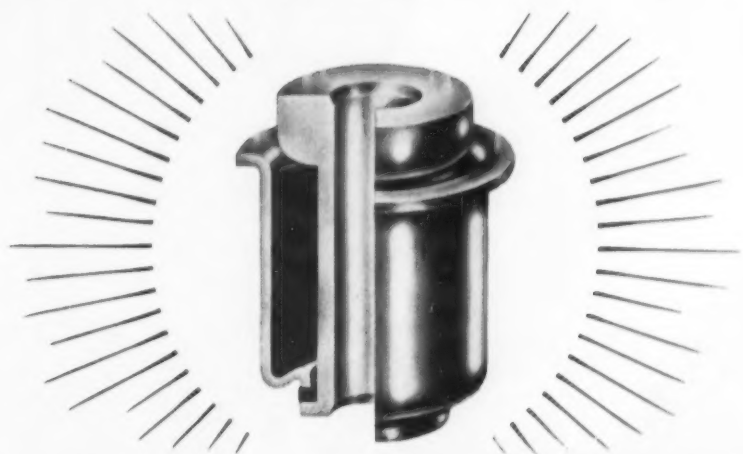
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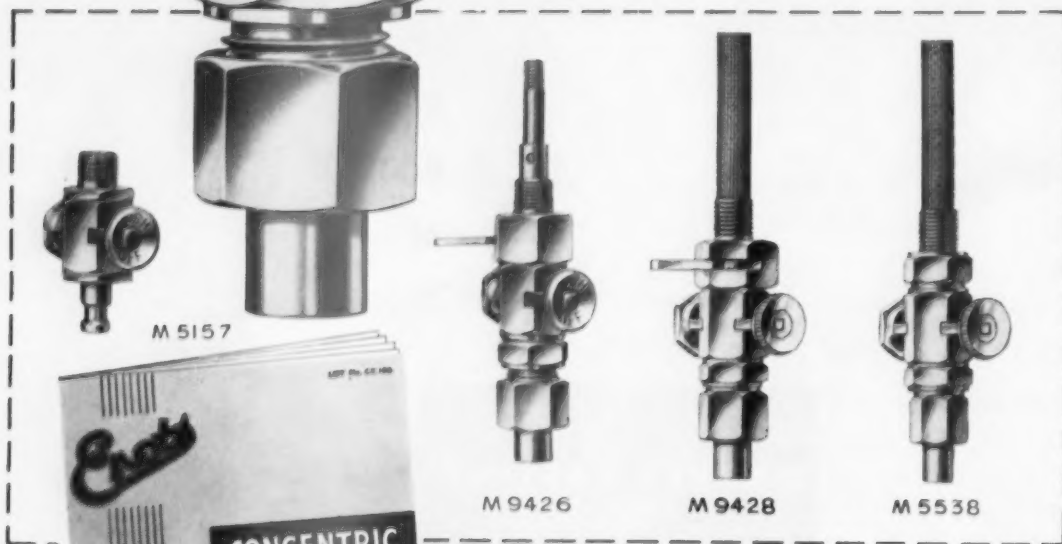
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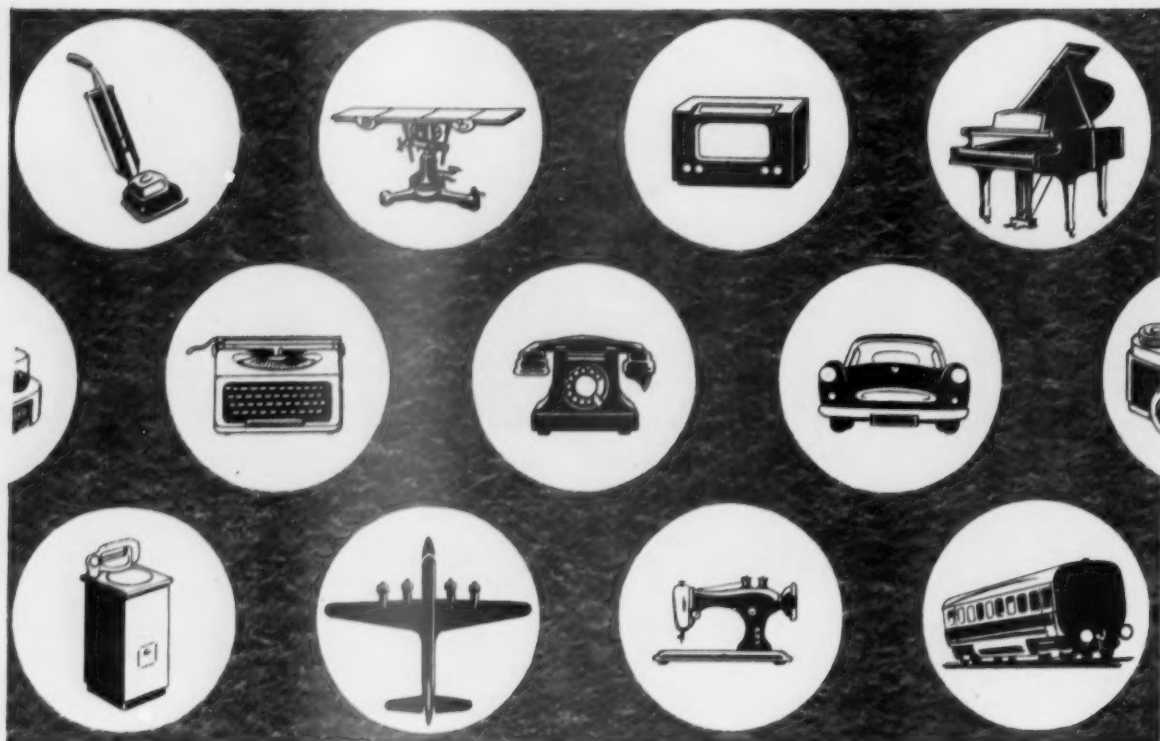
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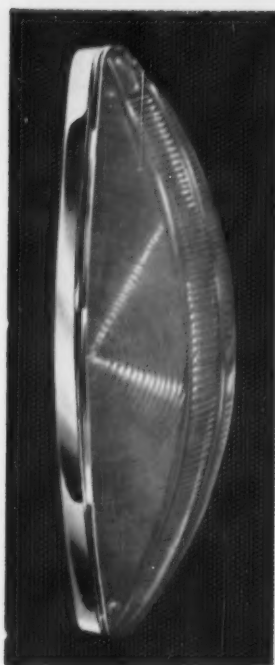
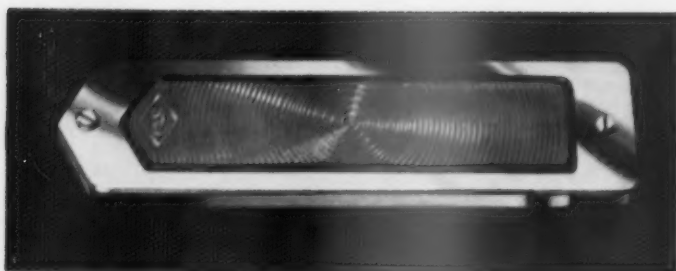


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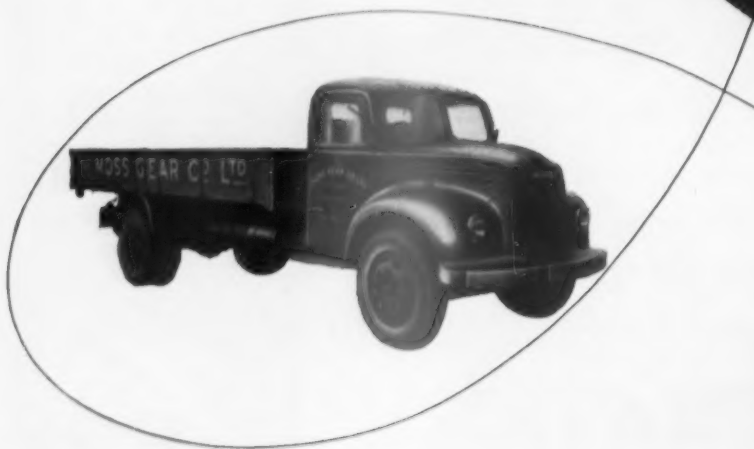
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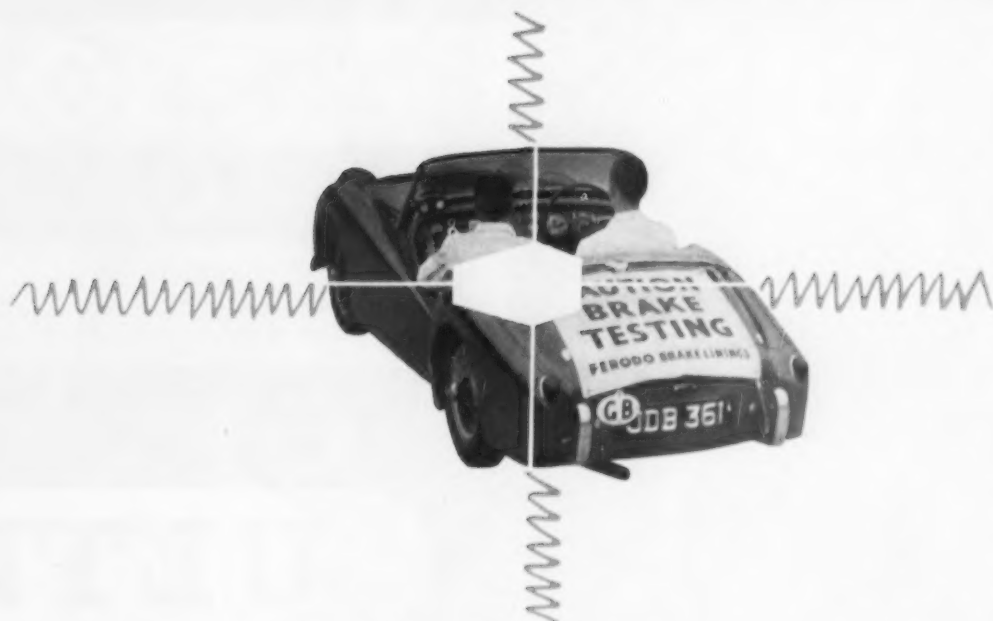
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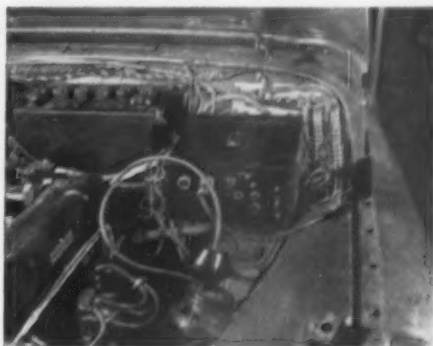
This car of the Ferodo Test Fleet is packed with equipment to record and analyze brake performance. For, when arranging schedules for tests, Ferodo research workers must know what a brake lining is expected to do under practical conditions, and that is what these instruments tell them.

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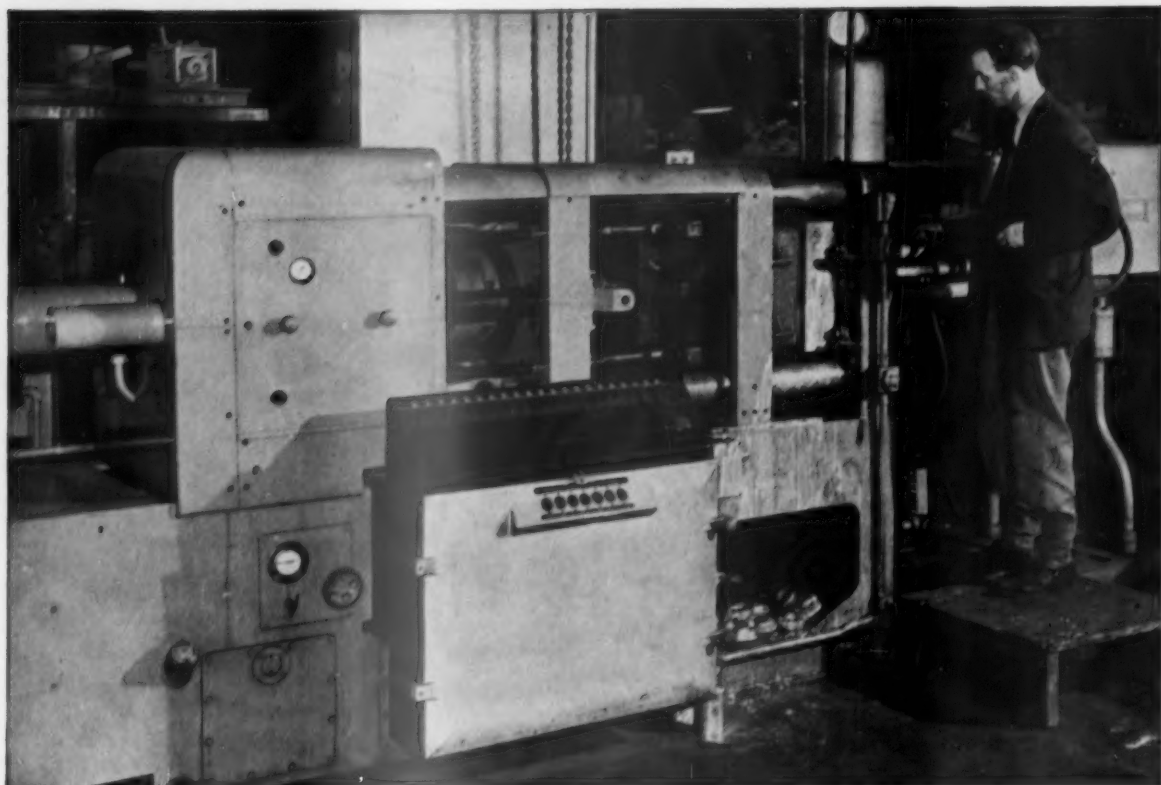
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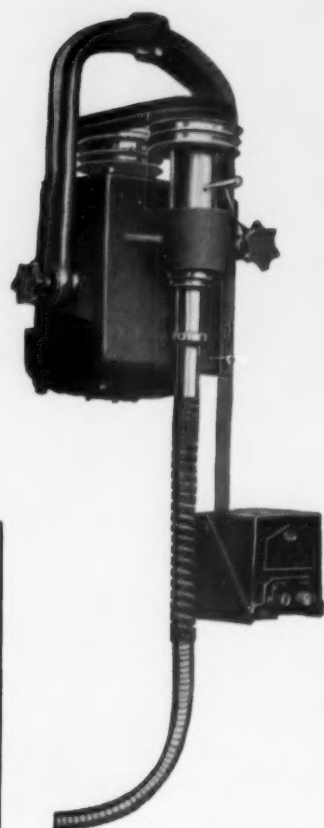
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**this is the liner
 that said
 goodbye to
 *brake squeal**

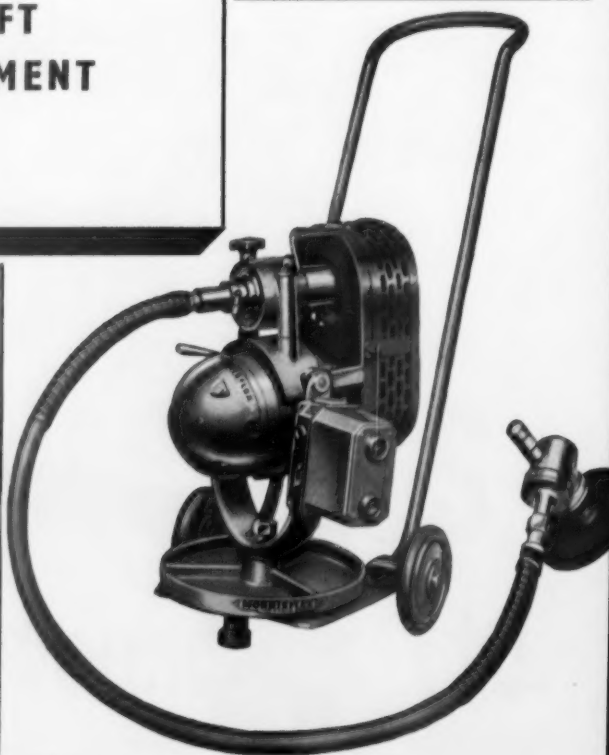


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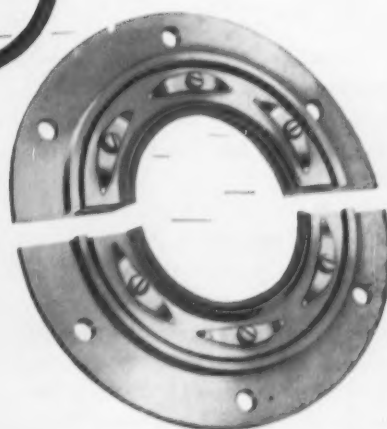
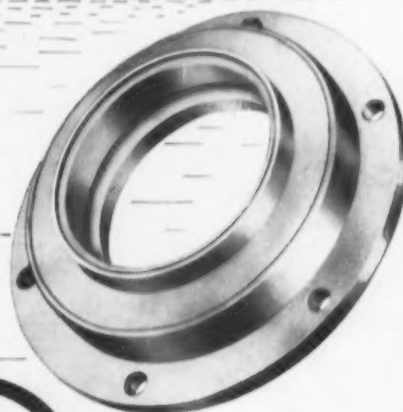
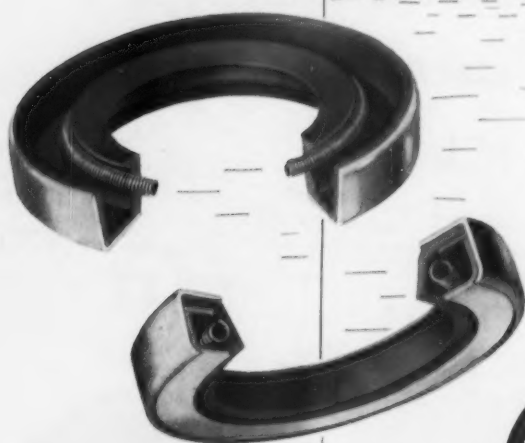


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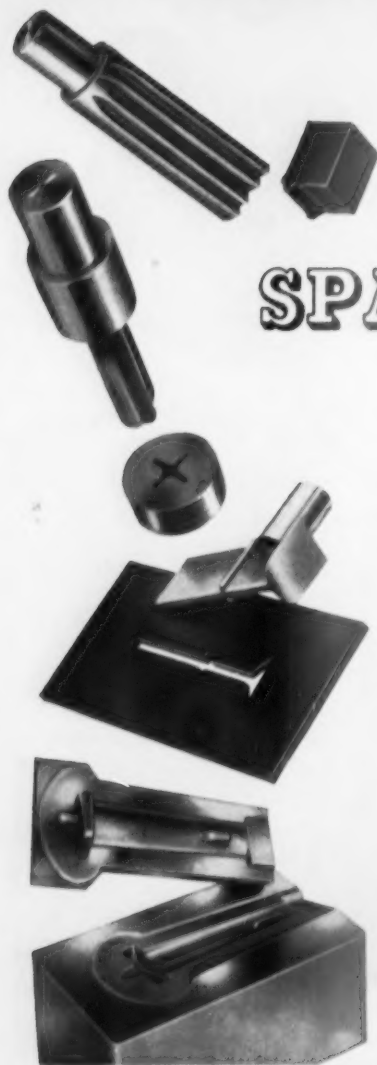
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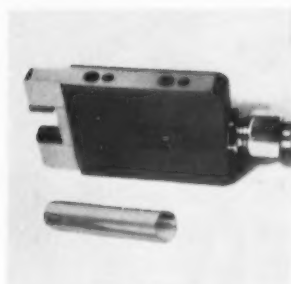
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AUTOMOBILE ENGINEER

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Editor J. B. Duncan

Editorial Staff T. K. Garrett, A.M.I.Mech.E. A.F.R.Ae.S. (Associate Editor)

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Free-Piston Engines

HITHERTO, little consideration has been given to the free-piston engine as the power unit for road vehicles, and developments have in the main been confined to large stationary installations and, more recently to ship propulsion. More recently still, Renault have built a locomotive with a free-piston unit which has done considerable successful service. They have also developed an experimental unit for road vehicle propulsion. In the United States of America too, developments are under way and on May 15th General Motors will introduce an experimental private car, the XP-500, with a free-piston engine.

In this country much development work has been carried out on the Pescara-type unit by Alan Muntz and Co. Ltd., and a short time ago they granted a licence to Associated British Engineering Ltd. for manufacturing Pescara-type free-piston engines. It is significant that the licence includes gasifiers with diesel pistons of less than 6in diameter. This suggests increasing interest in this type of engine for road vehicle work.

The free-piston engine is essentially an opposed-piston, two-stroke engine with large diameter air-compressor pistons arranged in tandem with the engine pistons. There are no connecting rods and no crankshaft. All the air compressed is used to scavenge and supercharge the engine cylinder to something in the order of 45 lb/in². The combustion products and the excess scavenge air are fed to a gas turbine which furnishes the useful power. Therefore, the complete power unit includes both reciprocating and rotary units. Generally, the integration of basically different principles results in a compromise, which, even if it is of a practical nature, is of lowered efficiency. However, as Dr. H. R. Ricardo states in reference to the compounding of the diesel engine with the gas turbine: "The piston engine is eminently suitable to deal with relatively small volumes at high pressure and temperature, and the turbine, by virtue of its high mechanical efficiency and large flow areas, to deal with large volumes at low pressures. Clearly the logical development is to combine the two in series to form a compound unit".

The free piston engine gasifier operates with compression pressures in the order of 1,000 lb/in² in the engine cylinder. This is the basic reason for its relatively high thermal efficiency of 40 to 45 per cent. Compounded with a gas turbine, the overall thermal efficiency is about 35 per cent, and a satisfactory rate of fuel consumption is attained.

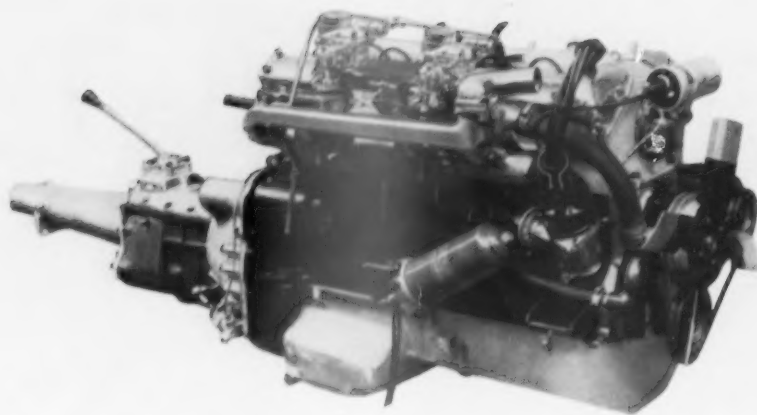
For road vehicles, the free-piston gasifier-turbine unit has attractive features. It is, of course, more suitable for large public service and commercial vehicles than for small private cars on the counts of first cost and complication. Such a power unit, because of its robustness, could prove especially useful for military vehicles.

At present, the gas turbine is generally looked upon as the development that may eventually oust the conventional piston-type engine, but, as is well appreciated, serious problems wait to be solved before the gas turbine can be commercially suitable for use in road vehicles. Apart from the manufacturing difficulties as regards both materials and methods, the specific rate of fuel consumption is inordinately high by piston engine standards. This can, of course, be countered by a system of heat recovery, but an adequate heat exchanger for a road vehicle has yet to be evolved. The recuperative type with transfer of heat through thin wall tubes is too bulky and vulnerable; the regenerative type in which heat is transferred in a moving matrix of refractory material has so far been impracticable as regards sealing.

There are not the same disadvantages with a free piston gasifier-turbine plant. Individually mounted and ducted together, the gasifier and turbine are easy to install. Furthermore, as the turbine operates at relatively low temperatures in comparison with those of a gas turbine, about 900 deg F, it can be of simple construction and of materials that are much less expensive and much more easily worked. The exhaust is relatively cool and there is no need for an expensive and bulky heat exchanger. The gasifier can be produced on simple machining equipment, and will use standard diesel fuel injection equipment.

As the turbine is purely rotary it cannot set up vibrations. The gasifier, although symmetrically opposed, is not completely vibrationless, owing to gas inertia, but it requires only light and simple mountings. In this respect it certainly has very great advantages over a conventional piston type engine.

Automobile engineers may be reluctant to accept the automatic spring plate valves of the air compressor, but these have been developed to a high degree of efficiency and in France they are operating satisfactorily in power stations generating electricity and in propulsion units for ships. There is no reason why they should not be equally successful in road vehicles.



The cylinder bore and the crankshaft bearing sizes in the Jaguar 2.4 litre unit are the same as in the XK range of engines. As a result, many of the components are common and the new unit is of exceptionally sturdy design

The Jaguar 2.4 Litre Engine

A High-Performance Unit designed to give scope for further development

IMMEDIATELY after the 1939-1945 war, Jaguar Cars Ltd. began to develop their new engines designed to replace the existing range and to outstrip all competitors so far as performance was concerned. Originally, it was intended to produce four- and six-cylinder units for installation in two sizes of vehicle to supersede the earlier models, with which the firm had built its reputation. With both sizes, considerable success had been experienced in meeting competition, and a strong following of enthusiastic owners had been established. However, the emphasis on export, particularly to dollar markets, led to the decision to abandon the idea of producing the four-cylinder version. Subsequently, entry into markets all over the world, and the need to reduce costs by expanding production, have made it desirable to produce a smaller saloon car in addition to the well known models powered by the XK range of engines. This new vehicle is powered by the six-cylinder 2.4 litre engine, introduced last October.

Although the four-cylinder layout has a slight advantage so far as cost and weight reduction are concerned, the six-cylinder type, because of its inherent smooth running characteristics, is considered to be more suitable for cars of the quality associated with the name of Jaguar. Moreover, six-cylinder engines are more acceptable than four-cylinder engines in the North American market. Another reason for the adoption of the six-cylinder engine was that integral construction was to be employed in the car to give the greatest possible structural efficiency, and thus to reduce weight and improve performance. With this form of construction, the prevention of the transmission of vibrations, such as those

due to the inherent secondary unbalance of four-cylinder engines, to the body is liable to be difficult.

To keep the prime cost to a minimum, it was decided that as many components as possible should be common to the XK and 2.4 litre ranges. Other incidental advantages have been obtained by following this policy of rationalization. One is that problems with regard to the supply of spares to all parts of the world are minimised. Also, the experience gained with the XK engines and the four-cylinder prototypes of the same basic design could be applied directly to the new engine.

In the new engine, the bore is the same as that of the XK unit, but the stroke is shorter and the connecting rod length has been shortened, so the ratio of connecting rod length: stroke is still 1.87 : 1. As a result, the height of the cylinder block has been reduced by 2½ in; this has considerably reduced the weight of the engine. The cylinder head is common to the XK and 2.4 litre units, as also are the crankshaft bearings. Because the bearings are of the same sizes as those used on the XK 140, and also because of the reduction in stroke, an exceptionally stiff crankshaft has been obtained. Evidently, further development to increase the speed and power output of the engine could be undertaken, should the manufacturers wish to do so later. In fact, the inlet valve lift is less than that of the XK, and the engine has been deliberately throttled at the carburettors and manifolds; therefore, the peak b.m.e.p. occurs at little over 2,000 r.p.m., whereas in the XK 140 it occurs at about 3,400 r.p.m.

The 2.4 litre engine has a stroke : bore ratio of 0.92 : 1, and its mean piston speed at maximum b.h.p. is 2,880 ft/min. The ratio of maximum torque : torque at maximum b.h.p. is 1.38 : 1, and the ratio of the speed at maximum torque : speed at maximum b.h.p. is 0.35 : 1. At 2,000 r.p.m., the maximum b.m.e.p. of 140 lb/in² is developed. The output in terms of b.h.p./in² piston area is 1.34, and 45 in terms of b.h.p./litre. Since the engine dry weight is 529 lb, the specific output is 0.212 b.h.p./lb. A minimum brake specific fuel consumption of 0.54 pt/b.h.p.-hr is obtained. The overall dimensions of the unit are as follows: height, less air filter, 27 in; width, less air filter and dip stick, 18½ in; length, less flywheel, 33½ in. It is installed at an angle of 4 deg 19 min 54 sec from the horizontal.

SPECIFICATION

Six cylinders. Bore and stroke 83 mm and 76.5 mm respectively. Swept volume 2,483 cm³. Maximum b.h.p. 112 at 5,750 r.p.m. Maximum b.m.e.p. and torque respectively, 140 lb/in² and 140 lb-ft at 2,000 r.p.m. Compression ratio 8 : 1 (7 : 1 optional). Forged, seven-bearing crankshaft, partly balanced. Overhead valves, actuated by two overhead camshafts. Twin solex downdraught B32-PB1-5/S carburettors, of the open-choke type, with 24 mm diameter choke. Hemispherical combustion chambers. S.U. electric fuel lift pump, type AUA 57.

Cylinder block and crankcase

A chromium iron cylinder block and crankcase casting is employed. Since the bore diameter and the spacing between the cylinder axes are the same as in the units of the XK range, it has been possible to design the block of the new engine so that most of the machining can be done, together with that of the blocks of the other units, on one production line. Simplicity of block design, a minimum of machining and relatively light weight are, of course, all inherent characteristics of the cylinder blocks for engines of the overhead camshaft layout. Further weight saving, as has already been mentioned, has been effected by the reduction in stroke and connecting rod length as compared with the XK range, but the crankcase, of course, is still inevitably the heaviest unit of the whole engine.

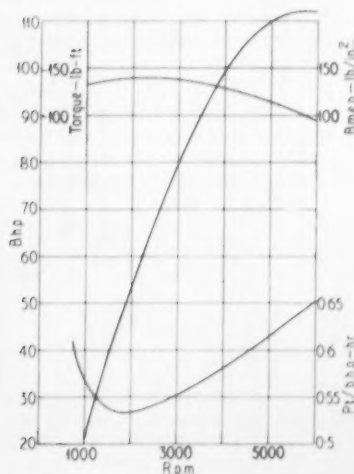
The cylinder bores are integral with the block and their wall thickness is $\frac{1}{16}$ in, while the minimum jacket space between the cylinders is $\frac{1}{16}$ in. They are hone finished to 25–35 micro-in. It has not been necessary to employ liners, because of the rate of wear experienced with the XK range has been exceptionally low. In fact, the manufacturer states that the need for rebore does not arise during the normal first-owner life. It is believed that the long service obtained between rebores is mainly due to freedom from distortion of the cylinders. This is, again, an inherent characteristic of the overhead camshaft layout, which allows the block design to be more or less symmetrical round the cylinder bores and also enables the cylinder head holding down studs to be favourably positioned and adequately supported. The support for the studs is afforded by vertical ribs between each stud boss and the lower deck of the block. These ribs form a continuation of each side of the transverse webs below the lower deck, which carry the main bearings. Because of this reinforcement, the thickness of the upper

and lower decks is only $\frac{1}{8}$ in. The employment of chromium plated rings in the top grooves of the piston also increase bore life. Tests have shown that the reduction in bore wear obtained by using these rings is at least 50 per cent.

Although the deep skirted type of crankcase design is probably the best in some respects for a four-cylinder engine, this manufacturer considers that for a six-cylinder unit it is better to have the sump face joint at the level of the axis of the crankshaft. This is because, whereas in the four-cylinder layout a deep skirt gives greater stiffness to react the vertical out-of-balance forces, six-cylinder engines are inherently well balanced and it is preferable to have the additional stiffness of the face joint flange at the level of the crankshaft axis, where it increases the lateral stiffness of the block. Moreover, the deep skirted type of crankcase is inevitably heavier than the short skirted type unless lateral stiffness is sacrificed. An additional advantage of the short skirt is the simplicity of the machining operations at the lower face of the block. In this engine, the seats for the main journal bearing caps are not countersunk in the lower face of the block, so lateral location of the caps is effected by two $\frac{1}{8}$ in diameter dowels, one on each side. To prevent the bearing caps from being fitted accidentally the wrong way round, their $\frac{1}{2}$ in diameter En 16T bolts are arranged eccentrically relative to the crankshaft axis, and each pair of dowels is offset to the rear of the transverse plane containing the axes of the bolts.

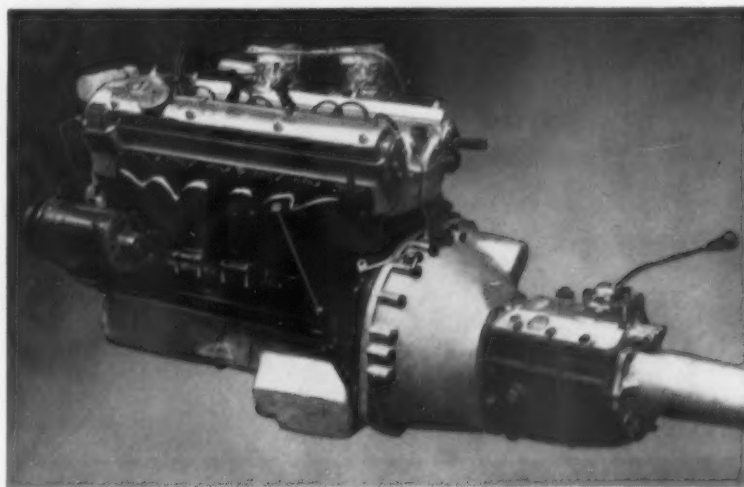
Crankshaft, connecting rods and pistons

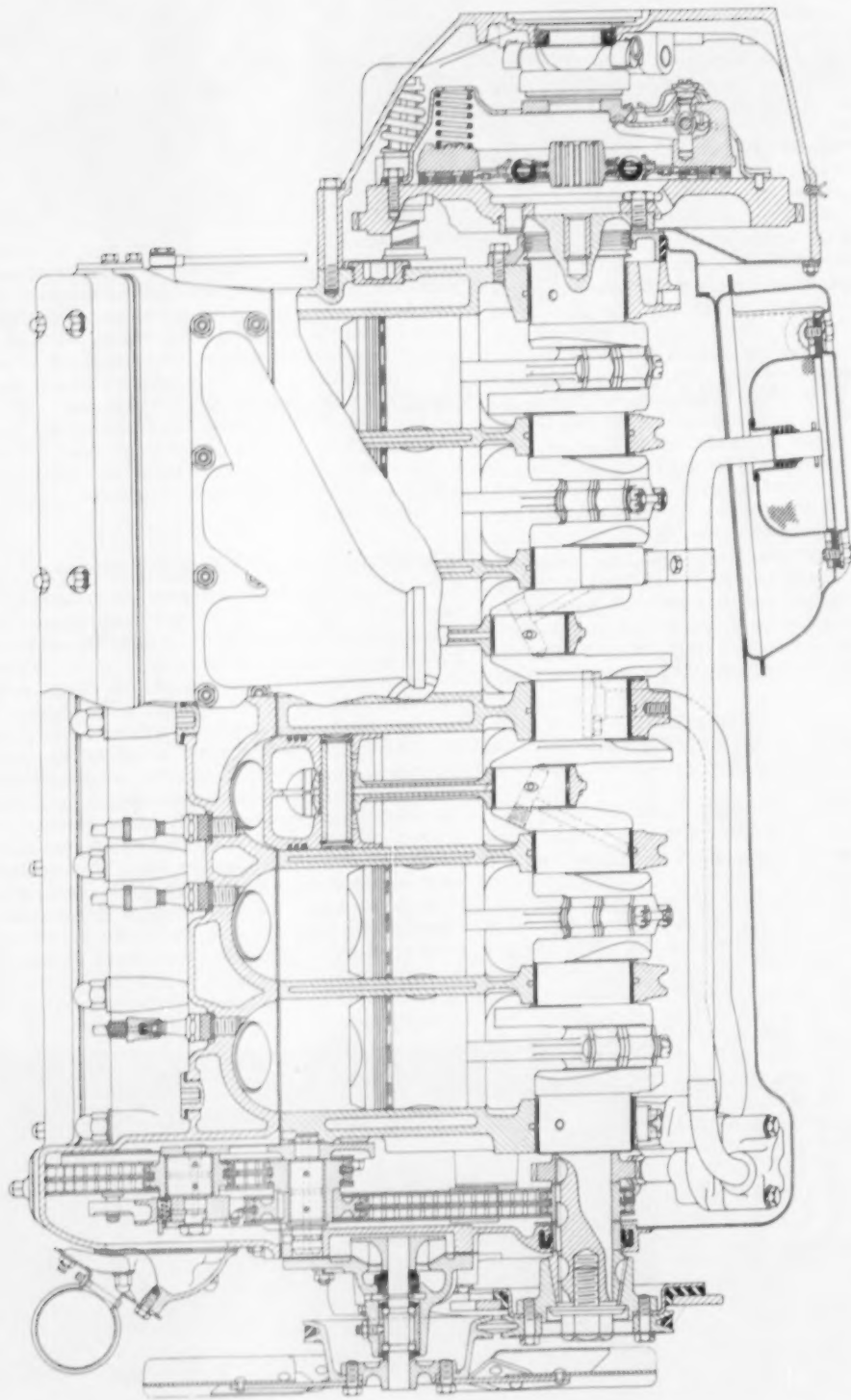
The En 16 crankshaft is of more or less conventional seven-bearing type of design. It is heat treated prior to machining to give a Brinell hardness figure of 270–295. A noteworthy feature is the large size of its bearings. The main journals are 2½ in diameter and their lengths are: front, centre and rear, 1½ in, remainder 1 in. Vandervell D2 Bimetal, babbitt-lined bearing shells are employed. Their diametral clearance is 0.0015–0.003 in and they are a 0.001 in nip fit in their housings. All the journals and pins are ground and hand lapped to a finish of about 10 micro-in. Axial location is effected at the centre main journal, where there are two semi-circular, babbitt-lined steel thrust washers, one in front of and the other behind the bearing cap. The edges of the oil holes, where they break through the surface, are stoned away to form an oval countersink. This reduces stress concentrations and acts as a lead-in for the oil. Also, if sharp edges are left round the holes, there is a danger of small



Above: The engine performance curves

Right: Clean exterior lines are characteristic of all the Jaguar engines, including this 2.4 litre unit





GENERAL ARRANGEMENT OF THE JAGUAR 2.4 LITRE ENGINE
Bore and stroke 83 mm x 76.5 mm. Swept volume 2,483 cm³

fragments of metal being broken off by the oil flow and damaging the bearings.

I-section connecting rods of En 16T3 are employed. They have a centre-to-centre length of 5½in. Their cross sectional dimensions are: width over flanges ½in and depth 1in. Each rod, complete with the bearing cap, shell and bolts, weighs 1 lb. 9½ oz.

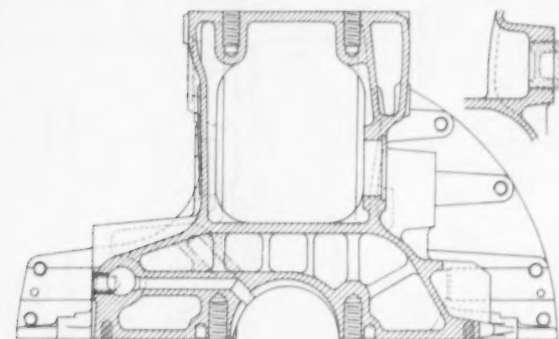
The big ends are split at an angle of 90 deg to the axis of the rod, but they are not too wide to be withdrawn through

hole at this end since there is only one radial hole in the small end bush, and this hole is in line with the lower portion of the duct in the web. The bush is of Clevite 10 phosphor bronze and is 1.08in long. Its interference fit in the eye of the rod is 0.0035–0.0045in, and the diametral clearance between it and the gudgeon pin is ± 0.0002 in.

An En 34 gudgeon pin, case hardened 0.020–0.030in deep to C57–C63 Rockwell is employed. It is of plain cylindrical section with an inside diameter of ½in and an outside diameter of ¾in. The surface finish is 4–5 micro-in. Seeger type circlips in grooves in the cross-holes axially locate the pin. The bearing length of the pin in each piston boss is ½in and the diametral clearance between the pin and the boss is from +0.0001 to –0.003in.

Brico, tin plated, AL.1 LX pistons of light construction are employed. Their weight, complete with gudgeon pins and rings, is 1 lb 3 oz 2 dm. They have flat crowns with chamfered edges to fit into the hemispherical cylinder heads and give a compression ratio of 8 : 1. A larger chamfer is machined on each side to clear the valves. The crown is stiffened internally by a single lateral web, and the loading is transmitted to the gudgeon pin bosses by two longitudinal ribs, one on each side of the bosses. A T-slotted skirt design, with the horizontal portion of the slot in the base of the oil ring groove, has been adopted. Three rings are fitted, two compression and one oil control, and all are above the gudgeon pin. Details of the rings are given in the accompanying Table.

The flywheel, which is spigoted over the flange on the rear end of the shaft, is an En 8 drop forging, and the 104-tooth starter ring gear is machined on its periphery. Its overall dimensions are 13.2in diameter \times 1½in thick and its weight and mass moment of inertia are 27 lb 10 oz and 5.1 lb-ft² respectively. Ten, ½in diameter high tensile steel bolts are used to secure it to the flange, and location is effected by two ½in diameter dowels. The reason for using this large number of bolts is that if the clamping effect of the bolts were inadequate to prevent relative movement between the shaft and flywheel, the dowels alone would react the whole of the inertia loading in excess of that taken by the friction between the clamped faces, until the metal round them deflected sufficiently for the clearance in the bolt holes to be taken up. Only then could the bolts begin to take a share of the shear load.

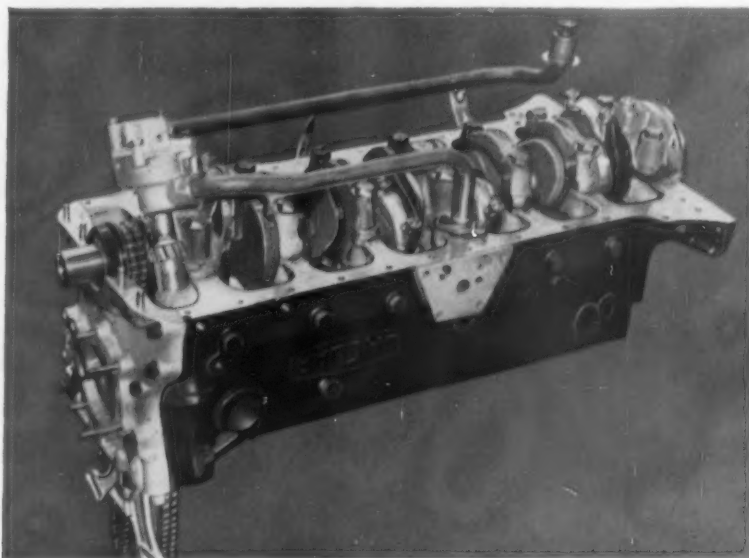


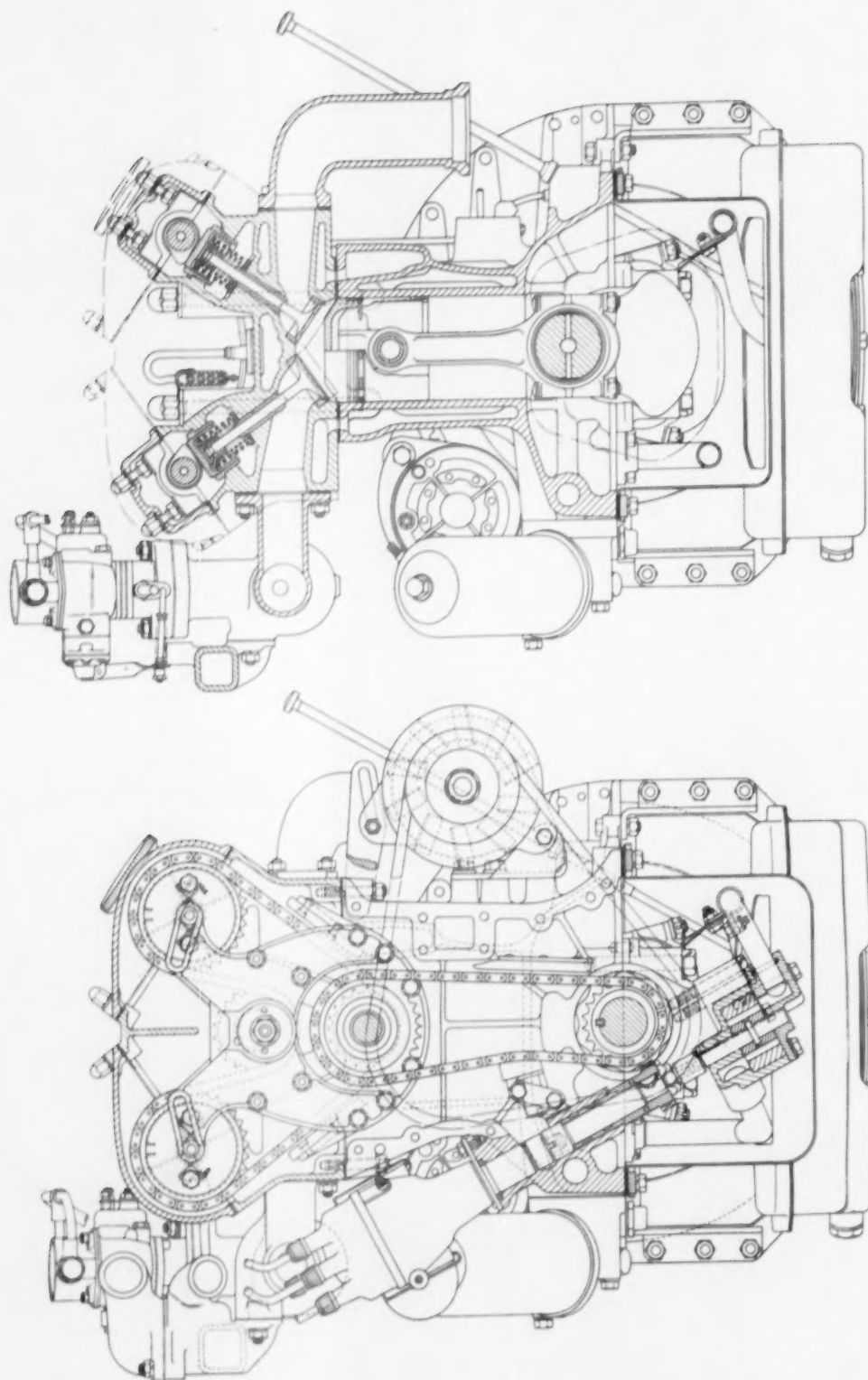
Cross section of the cylinder block and crankcase casting, showing how the cylinder head holding down studs are supported. The scrap view illustrates the arrangement for installing an electric heater unit in the jacket, for countries with very cold climates

the bores of the cylinder. Each cap is located by the fit of its two ½in diameter split-pinned En 24T retaining bolts. The shanks of the bolts are not waisted. Vandervell D2 Bimetal, white metal lined, steel shell type bearings are employed. They are 1in long and the crank pin diameter is 2.0863in. The diametral clearance is 0.0009–0.0025in and the nip of the bearing shells in the bearing housing is 0.001in.

A ½in diameter radial hole in the upper half of the bearing shell communicates with a ¾in diameter duct drilled axially through the web of the rod. Although the duct breaks out at the top end of the rod, the oil cannot escape through the

Inherently good features of the cylinder blocks for engines of the overhead valve layout are simplicity of design, a minimum of machining and relatively light weight. Despite the difference in stroke between the 2.4 litre and XK engines, a considerable amount of rationalization has been effected, so far as the machining of the cylinder block and crankcase unit is concerned





CROSS SECTIONS OF THE JAGUAR 2.4 LITRE ENGINE

Relatively small half-speed wheels are used, because a two-stage, two strand chain drive has been adopted for the timing drive. The main journal bearing caps are located by dowels and their holding down bolts are offset to prevent incorrect assembly.

The oil seal immediately in front of the flywheel is a return scroll machined round the crankshaft. This scroll is surrounded by a two-piece housing of cast aluminium. The upper half of the housing is bolted to the rear face of the crankcase, while the lower half is bolted to the upper one. An annular groove is machined in the bore of this housing and radial holes in the lower half drain oil from it, through the drainage space for the oil thrower in front of the scroll, to the sump. This arrangement is incorporated to deal with excessive quantities of lubricant that might suddenly flood the return scroll during violent acceleration of the vehicle. The rear end of the sump has a lipped, semi-circular cut-out in it, and the lip is pulled up against a rubber bonded cork strip in a groove round the lower half of the housing. A

PISTON RING DATA

	Top	Second
Compression rings:		
Type	Plain chromium plated	Plain oxidized
Gap	0.020-0.015in butt joint	
Face width	0.0787-0.0777in	
Radial thickness	0.130-0.124in	
Depth of groove in piston	0.136-0.131in	
Side clearance	0.003-0.001in	
Oil control ring:		
Type	Slotted scraper ring	
Gap	0.016-0.011in butt joint	
Face width	0.156-0.155in	
Radial thickness	0.127-0.119in	
Depth of groove in piston	0.136-0.131in	
Side clearance	0.003-0.001in	



Spider assembly carrying the timing drive chains and the intermediate and eccentric jockey sprockets

collar round the shaft between the return grooving and the rear journal bearing forms the oil thrower and works in a drainage space in the front end of the housing round the return grooves. Oil flung off the thrower falls to the bottom of the space and drains away through a hole drilled in the base of the bearing cap.

The shaft is only partly counterweighted to oppose the inertia loads of numbers 3 and 4 reciprocating masses. This is a compromise, because the addition of counterweights large enough to balance the loads fully would have introduced formidable problems with regard to torsional vibration. The counterweights are dynamically balanced by two more weights, one on number 2 web and the other on number 11 web. Particular care is exercised with regard to the balance of the complete assembly. First the shaft is dynamically balanced on an Avery machine. Next, the flywheel, which previously has been checked on a Micropoise Static Balancer, is fitted; the whole assembly is then corrected statically. A final check is made after the clutch has been bolted on.

The torsional vibration damper is a Metalastik rubber-to-metal bonded, inertia-ring type unit. It is spigoted, together with the fan belt pulley, on to a flanged, steel hub. This hub has a tapered bore and is pulled up on to a split cone on the front end of the crankshaft. The cone is keyed on to the shaft, and the hub is keyed to the cone. This arrangement is employed, of course, to ensure that there is no possibility of relative movement between the shaft and the hub of the damper.

Timing drive

At the front end of the crankshaft the extension is $4\frac{1}{2}$ in long \times 1 $\frac{1}{2}$ in diameter. Assembled on to it in the following order are: the gear, of Fox 709 steel, for driving the oil pump and distributor, the En 8D(R) timing chain sprocket, a large diameter plain washer that acts as an oil thrower, a distance tube and the split cone that carries the 5in overall

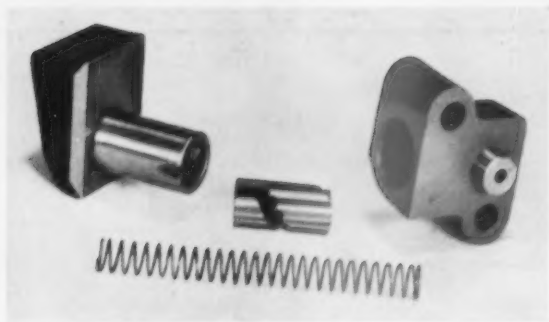
diameter aluminium, fan belt pulley, and torsional vibration damper assembly. The whole assembly is retained by a $\frac{3}{8}$ in thick washer and a $\frac{1}{2}$ in diameter set bolt screwed axially into the front end of the crankshaft extension. Of the three Woodruff keys transmitting the drive to the components, that at the rear drives the oil pump gear, and that at the front drives the damper cone, while the centre one, which drives the chain sprocket, also projects about $\frac{1}{8}$ in forward of the sprocket into keyways in the thrower disc and the distance tube. A lip type oil seal bears on the distance tube. It is carried in a channel section housing in the front cover of the cell at the forward end of the crankcase, in which the timing chain is enclosed.

The upper portion of the timing case is formed by a cell cored in the front end of the cylinder head casting. There are three bolted-on covers on the cell, one above each half speed wheel and one on the front. The two-stage form of chain drive adopted has the disadvantage that it increases the overall length of the engine, but it has a number of marked advantages. Not least among these is the fact that some of the speed reduction can be made at the intermediate sprocket. Therefore, much smaller diameter camshaft sprockets can be employed and so the timing case does not have to be so wide as with the single-chain layout.

Two $\frac{1}{2}$ in pitch, double-row chains are employed: one transmits the drive from the crankshaft sprocket to an idler, whence the other takes the drive on to the two half speed wheels. The idler comprises two sprockets, one keyed on to a forward extension of the boss of the other. This double sprocket assembly has a phosphor bronze bush pressed into its central boss, and is carried on a spindle, the



Front cover removed to show the timing drive and chain tensioning arrangements



The components of the Renold hydraulic chain-tensioner

axis of which is $8\frac{1}{2}$ in above that of the crankshaft. The centre-to-centre spacing between it and each half speed wheel is $6\frac{1}{2}$ in. A noteworthy feature of the assembly is that the sprockets are at the ends of the arms of a Y-shaped spider, bolted to the front wall of the cylinder block. This spider is used to carry the half speed wheels and their chains when the cylinder head or camshafts are removed, so that servicing operations can be performed on the head without disturbing the valve timing.

The central portion of the spindle on which the intermediate sprocket assembly is carried is $\frac{1}{2}$ in diameter. Its front end is reduced to $\frac{1}{4}$ in diameter where it is carried in a hole in the front cover, and its rear end is reduced to $\frac{1}{2}$ in diameter. The rear end projects through a hole in the back

projects through the hole in which it is carried in the spider, and a flat is machined on its periphery. This flat locates a large diameter washer, which has a corresponding flat in its bore, to register against that on the spindle; thus, rotation of the washer also rotates the eccentric. The whole assembly is retained by a nut, which is screwed on to the front end of the spindle.

To lock the setting of the eccentric, the periphery of the washer is serrated and located against rotation by a spring-loaded plunger in a blind hole in the spider. This plunger has a flat machined at its front end. The serrations that lock the setting are cut on the flat and then their front ends are machined off so that only a short length remains to engage those round the washer. Thus, to adjust the setting of the eccentric, it is not necessary to undo the nut on the front end of the spindle: instead, the plunger is pushed into its housing until the serrations disengage and the washer is in line with the flat in front of the serrations on the plunger. Then the washer, and with it the eccentric, can be rotated to effect the adjustment. Finally, it is locked again by releasing the plunger, which springs out so that the serrations engage. Access to make this adjustment is gained by removing the cover on the front of the cell in which the timing gear is housed.

A Renold hydraulically-actuated, slipper type tensioner takes up the slack in the lower chain, and a nylon-faced, fixed slipper steadies the drive side. Both the tensioner and the fixed slipper, of course, bear on the outer face of the chain. The hydraulic tensioner has the advantage that it also serves as a chain lubricator. It comprises a body casting drilled to receive the hollow cylindrical stem of a Neoprene-faced slipper. In this stem is a spring-loaded plunger. The

A view of the cylinder head, with the camshaft covers removed, illustrating the simplicity of the overhead camshaft design from the point of view of both layout and assembly of the valve actuating gear on the head

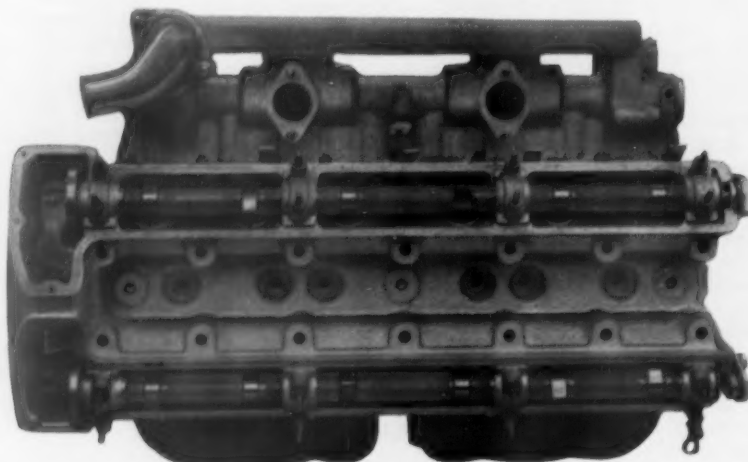


plate of the spider and is spigoted into another hole in the front wall of the cylinder block. The hole in the block locates the spindle relative to the crankshaft. Axial location of the spindle is effected by a circlip in a groove round its rear end; this circlip seats in a shallow counterbore round the hole in the cylinder block, in which it is retained by the back plate of the spider. The sprocket assembly is located between the back plate and a collar formed round the forward end of the large diameter central portion of the spindle.

Directly above the intermediate spindle is the eccentric jockey sprocket for the upper chain. This sprocket is mounted between the spider and its back plate. The eccentric is $1\frac{1}{8}$ in diameter and the ends of the spindle, where they are carried in the spider and back plate, are turned down to $\frac{1}{2}$ in diameter. At the front end, the spindle

whole assembly is bolted to the front wall of the crankcase, with the axis of the stem of the slipper normal to the strand of the chain.

The coil spring, which seats at one end in the piston-shaped plunger and at the other in the closed outer end of the stem of the slipper, is in compression so that it forces the slipper against the chain. When the engine is running, lubricating oil from the main gallery passes through a metering hole into the body of the tensioner and, since it is under pressure, adds to the force of the spring in pushing the slipper against the chain. There is also a bleed hole, drilled axially through the outer end of the stem of the slipper and through the Neoprene facing, to lubricate the chain.

The plunger is simply a ratchet device to limit the motion of the slipper assembly away from the chain. In its side

there is a spiral slot in which is registered a peg projecting radially inwards in the end of the bore of the hollow plunger. One edge of this slot is serrated in such a manner that if the slipper is forced inwards against the hydraulic pressure and the spring, the peg moves until it seats in the nearest serration so that further movement is prevented. As extension of the chain takes place, the stem is pushed outwards stage by stage and the peg, sliding in the spiral groove, rotates the plunger. This brings fresh serrations, one after the other, in line with the peg and thus locks the plunger progressively further and further out. Tension in the upper chain is adjusted by the eccentric jockey sprocket meshing against the outer face of the strand between the two camshaft sprockets. On each flank of the chain, a nylon-faced brass slipper, bolted to the cylinder block, steadies the sections between the intermediate and camshaft sprockets.

Each camshaft sprocket is an En 8D(R) ring spigoted on to a flange at the front end of the camshaft. The teeth are machined on the outer periphery, and there is an internally serrated flange in the bore. A disc with serrations on its periphery, to engage those in the internal flange, is spot welded to a retainer plate, and the whole assembly is secured to the camshaft by two set bolts. Thus, the retainer plate clamps the internal flange of the sprocket against the front face of the flange at the forward end of the camshaft. A circlip in a groove in front of the retainer in the sprocket prevents the serrations from sliding out of engagement when the assembly is withdrawn from the camshaft flange. There are 131 serrations; this makes possible an adjustment of as little as 1 deg 22 min to the timing of the valves. A short shouldered spindle is spigoted into a hole in the centre of this plate assembly and is retained by a Belleville washer and a circlip in a groove round its rear end. The front end of the spindle is threaded and reduced in diameter to project through a slotted hole in the arm of the spider to which, if the camshaft is removed, the spindle and plate assembly is clamped by a slave nut. Thus, the sprocket is free to rotate if for any reason the engine has to be turned while the head is off.

If a camshaft has to be removed, the two set bolts that secure the sprocket to the shaft are unscrewed and a slave nut is assembled on to the end of the central spindle to withdraw the sprocket and clamp it on to the end of the spider arm. Then the shaft can be taken away and replaced again without altering the setting of the valve timing. When the camshafts are replaced, the crankshaft is first set so that No. 1 piston is at T.D.C. Then the camshaft setting relative to the crankshaft is fixed by means of a special gauge. This gauge is located on the upper joint face on the head and has a tongue that registers in a slot in the flange round the camshaft, immediately behind its front bearing. If the whole of the head assembly has to be removed, both camshaft sprockets have to be withdrawn in the same way.

However, in these circumstances, they must be slid towards each other along the slots in the spider arms; otherwise, when the assembly is removed, they will not clear the sides of the cell in the head.

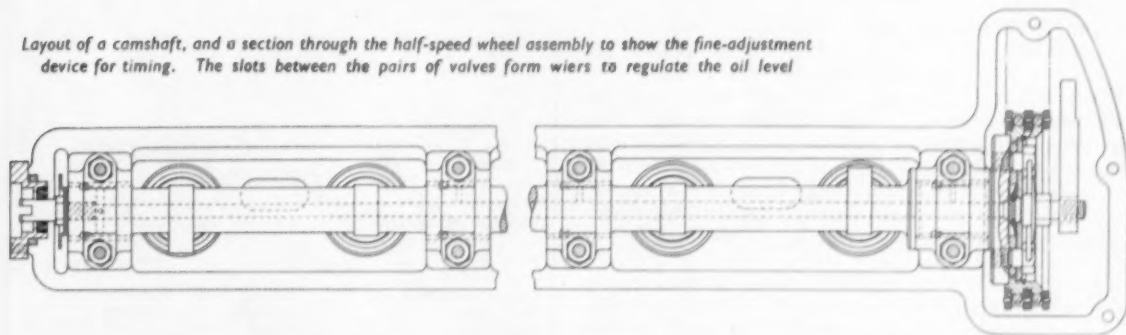
Camshaft and valve gear

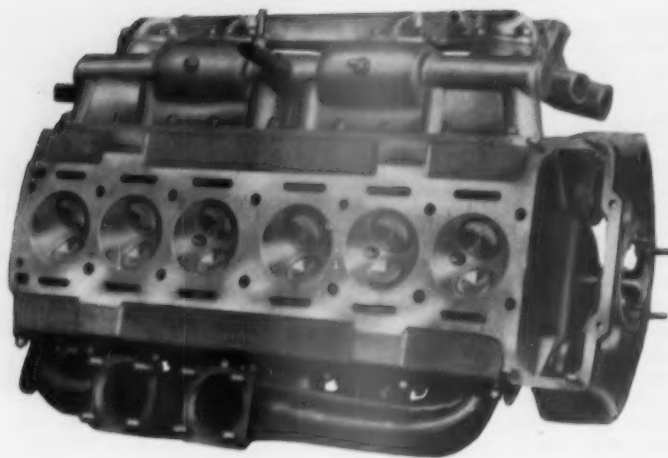
The advantages of the twin overhead camshaft layout adopted are as follows. Since there are no rockers, and the side thrust from the cams is taken by the tappets interposed between them and the valve stems, the rate of wear of the valve stems and guides is practically zero. This is a noteworthy feature because it eliminates the possibility that wear will cause excessive clearances to develop between the stems and guides. Excessive clearance particularly on the exhaust side, of course, impedes the flow of heat from the valve, and as a consequence, overheating occurs and tends to burn the relatively large amounts of oil that get into these big clearances. The resultant accumulation of carbon, and the side thrust on the valve stems, is liable to cause valve sticking—a trouble which the Company say is unheard of on these engines. Another feature that helps to prevent the passage of too much oil down the stems is the employment of piston type tappets over the upper ends of the valve and spring assemblies.

Adjustment of the tappet clearances is effected by interposing hardened En 24Z steel washers of appropriate thicknesses between the tappets and the valve stem ends. In service, this is done as follows. First, the existing clearances at each tappet are checked and recorded. Then the camshaft is removed and the thicknesses of the discs between the tappet and valve stem ends are measured. By comparing these measurements with the recorded clearances, it is a simple matter to calculate the thicknesses of the replacement discs that must be inserted to correct the clearance. After the new discs have been fitted, the camshafts are reassembled in the head. The whole sequence of operations can be done either *in situ* or with the head removed and placed on a bench.

The reason why this disc and tappet arrangement can be employed is that adjustment is necessary only at infrequent intervals. This is because with the twin overhead camshaft layout, the total weight of the reciprocating components of the valve train is low and the valve spring load is light. Therefore, for any given speed, the loading of the rubbing surfaces of the cams and tappets is only about half that obtained with the conventional layout comprising a low camshaft, push rods and rockers. Moreover, since there is no sliding between the ends of the valve stems and the tappets, except that caused by the inevitable small amount of tappet rotation, the wear of their contacting surfaces is negligible. Another advantage of the employment of lightly-loaded springs is that there is no danger of seat wear or valve breakage.

Layout of a camshaft, and a section through the half-speed wheel assembly to show the fine-adjustment device for timing. The slots between the pairs of valves form wiers to regulate the oil level





The arrangement of the combustion chambers, and inlet and exhaust manifolds

Four, Vandervell D2 Bimetal, white metal lined, steel shell type, plain bearings carry each Monikrom cast iron camshaft. All are $1\frac{1}{8}$ in long \times 1 in bore and the diametral clearance is 0.0005–0.002 in. They are carried in housings machined directly in the cylinder head casting and are retained by caps secured by two $\frac{5}{16}$ in diameter studs. Extensions of these studs also secure the cast aluminium alloy cover on top of the camshaft housing. The interference fit between the bearing shells and their housings is 0.001 in. Axial location is effected at the front end, where the bearing housing is between two flanges round the shaft. Both shafts are $\frac{3}{8}$ in diameter between the cams and they have $\frac{1}{8}$ in diameter holes drilled axially through them. Radial holes, $\frac{3}{16}$ in diameter, feed oil to the bearings.

The lift of the inlet cam is low, by comparison with that in the XK range; this is to restrict the breathing. On the inlet side, the lift is $\frac{1}{8}$ in, and the nominal period is 240 deg. At 6,000 r.p.m., the maximum positive acceleration of the tappet, on the flank of the cam, is 20,800 ft/sec² and the maximum negative acceleration, on the nose, is 6,400 ft/sec². On the exhaust camshaft, the lift is also $\frac{1}{8}$ in, but the nominal period is 252 deg. A maximum positive acceleration of 30,900 ft/sec² is obtained and the maximum negative acceleration is 5,620 ft/sec². The cam profiles are chilled to a Rockwell hardness figure of C50–51.

The valves are set at an included angle of 70 deg, the axes of each pair being in the same transverse vertical plane as the axis of the cylinder that they serve. Their material specifications are given in the accompanying Table. The tappets are of chilled cast iron. They operate in flanged guides pressed into housings in the head. These guides are of austenitic iron because the high coefficient of expansion of this material helps to maintain the interference fit between them and their housings in the aluminium alloy head. There is a diametral clearance of 0.0019–0.0008 in between the tappets and the $1\frac{1}{8}$ in diameter bores of their guides.

Each valve has two springs, which are located radially by their shouldered retaining washers. Split tapered collets lock the assembly to the valve stem in the normal manner, and the hardened steel disc that is interposed between the end of the stem and the tappet is located radially in a counter-bore in the upper end of the retaining washer. A circular, dished steel pressing forms the seating for the lower end of each pair of valve springs.

Cast iron valve guides are fitted. They are of a simple cylindrical section that can be finished by centreless grinding. The inlet and exhaust guides are not interchangeable because their lengths are different. At the lower ends of the guides,

only the outer periphery is chamfered to remove the burrs without appreciably reducing the cross section. On the other hand, the upper end of each guide is completely chamfered to form a knife edge round the bore. This knife edge scrapes most of the oil film off the stem and prevents an excessive quantity of lubricant from passing into the guides. At this end, the reduction of cross section, due to the large chamfer, does not matter because, unlike the lower end, it is not subject to high temperatures. Before the guides are assembled into their housings, the head is heated to 80 deg C, so that the aluminium is not scored during the pressing-in operation.

Both the inlet and exhaust valves are of the modified tulip type, which gives a favourable shape to the gas passage in the port, and therefore good flow characteristics. Also, when the valves are running hot under heavy loading, this type retains its seat shape better than the flat-headed valve. Heat

dissipation evidently has not been a serious problem, probably owing to the use of an aluminium head. A 30 deg seat angle has been adopted for the inlet valve, while the angle of the exhaust valve seat is 45 deg. This arrangement is considered to offer the best compromise, since it meets the requirements for smooth flow through the inlet valves and good seating of the exhaust valves to prevent blow-by. Both

VALVE DATA

Material	Inlet	Exhaust
	Silicon chrome steel En 52	Austenitic steel Fox 1282
Head diameter	$1\frac{1}{8}$ in	$1\frac{7}{8}$ in
Throat diameter	$1\frac{1}{8}$ in	$1\frac{1}{8}$ in
Stem diameter	$\frac{5}{16}$ in	$\frac{5}{16}$ in
Diametral clearance in guide	0.004	0.002 in
Seat angle	30 deg	45 deg
Face width;		
on valve	0.1538 in	0.142 in
on seat	0.076 in	0.1989 in
Seat material	Brimol centrifugally cast	
Spring material	Spring steel En 49D	
Spring rate;		
inner	69.3 lb/in	
outer	77.4 lb/in	
Spring length, free;		
inner	1.515 in + 0.020 in	—0.000 in
outer	1.775 in + 0.055 in	—0.000 in
Spring length, installed;		
inner	$1\frac{7}{8}$ in	
outer	$1\frac{1}{8}$ in	
Spring surge frequency;		
inner	24,200 c/min	
outer	19,560 c/min	
Number of coils;		
inner	6	
outer	5	
Coil diameter;		
inner	0.74 in	
outer	1 in	
Wire gauge;		
inner	12 S.W.G. (0.104 in)	
outer	10 S.W.G. (0.128 in)	
Valve lift	$\frac{1}{8}$ in	$\frac{1}{8}$ in
Valve crash speed	7,000 r.p.m.	
Valve guide material	Cast iron bar 2K11 or NC	
Valve guide length	$1\frac{1}{8}$ in	$1\frac{1}{8}$ in
Valve guide inside and outside diameters	$\frac{1}{8}$ in	$\frac{1}{8}$ in
Tappet clearance	0.004 in cold	0.006 in cold
Valve opens	10 deg B.T.D.C.	57 deg B.B.D.C.
Valve closes	50 deg A.B.D.C.	15 deg A.T.D.C.

valve stems are undercut; the undercut on the inlet valve is 0.031in on the diameter and extends up to a level about $\frac{1}{16}$ in below the lower end of the guide; that on the exhaust is 0.020in on the diameter and extends to about $\frac{1}{16}$ in above the lower end of the guide. The object of the undercut is to form a scraping edge to keep the lower ends of the guides free from carbon.

Brimol austenitic cast iron valve seats are fitted. This material, which has a high percentage nickel content, has a coefficient of thermal expansion of 0.000018/deg C, as compared with that of 0.000012/deg C for a straight cast iron, and 0.000022/deg C for D.T.D. 424. Although the cost of austenitic iron is greater than that of a straight iron, its high coefficient of expansion is a material advantage for seats to be fitted in an aluminium head. Before the seats are fitted, the head is preheated to a temperature of 150 deg C. The pre-heating both for fitting the seats and the guides is done in a controlled muffle furnace. A simpler method of fitting these components would have been to freeze them and insert them into the head without preheating it, but this is impracticable because a change in the properties of austenitic iron takes place at low temperatures. Although the exhaust valve seats are of plain rectangular section, with an internal chamfer machined in each to form a seating face, the bores of the inlet seats are shaped to form a continuation of the valve throat profile, and thus to give efficient breathing.

Cylinder head

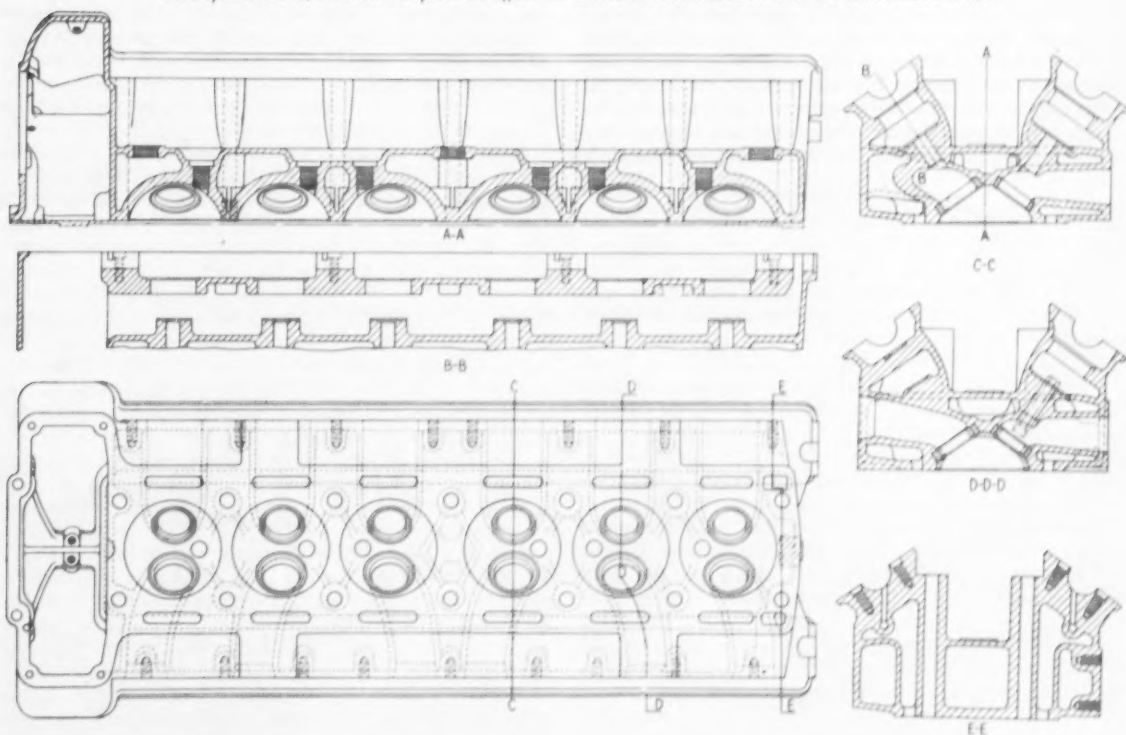
An aluminium alloy head has been used for two main reasons. One is its high thermal conductivity and the other is its light weight. The complete head assembly weighs 50 lb, whereas had the casting been of iron, the weight would have been about 120 lb. Other advantages are the machinability of the aluminium and the ease with which the head can be handled at all stages of the production. A disadvantage is the extra care needed to avoid damaging the component when it is being handled on the production line.

However, this can be largely overcome by taking a cut of 0.020in from the joint face after all other operations on the head have been completed. Before it was finally decided to use D.T.D. 424, a low expansion alloy was tried, but was rejected because of its high cost and not entirely satisfactory machining characteristics. The slightly lower rate of conductivity of this alloy was not considered to be of major importance. There has never been any sign of the seats working loose, even in the most highly rated XK units, so the decision to use D.T.D. 424 is justified.

The design of a cylinder head to be made of aluminium calls for the exercise of a certain amount of skill to avoid distortion. An important requirement is that the metal round the valve seats and guides shall be symmetrically disposed. Another is that the internal ribbing and the holding down bolts shall be suitably positioned. The tendency for distortion to occur arises from two main causes. Of these, the first is the load applied to the head by the holding down bolts, and the second is the differential rates of expansion between the relatively large and continuous mass of metal immediately beneath the camshafts, and the light arched sections of the combustion chambers and lower joint face.

Hemispherical combustion chambers are employed. This form of chamber is undoubtedly the best if a high b.m.e.p. is the most important requirement. Its advantages are as follows. Large diameter valve seats can be accommodated and, because of the tendency of the hemisphere to blend with the contours of the valve throat, the whole of the annular area between the head and the seat when the valve is open can be made effective; therefore, the gas flow is as good as can possibly be obtained. An unimpeded flow of cooling water to the exhaust valve seats is obtainable because of the favourable relationship between the ports and the jacket walls. Machining is simple; for this engine, a single form-cutter rotating about a fixed centre is employed. If the turbulence is properly controlled, and the sparking plug

In the cylinder head, the induction ports are offset and curved so as to induce swirl in the combustion chamber



is reasonably near the centre, the combustion characteristics of the hemispherical head are the best obtainable.

Pre-ignition and running-on have not been experienced with this combustion chamber. Some authorities maintain that the hemispherical head is inherently free from these troubles. Others* hold that running-on is a function of charge temperature; the reasons they give are as follows. If an engine running at its working temperature is switched off, the gas flow velocity in its induction manifold falls steadily, so that the heat picked up from the manifold and its jacket or hot-spot is greater; the fall in water pump delivery reduces the rate of thermal transfer from the cylinder head to the coolant; more time is available for pre-combustion reactions and for the charge to be heated by combustion chamber hot-spots. It is possible that the reason why the Jaguar is free from running-on troubles is that its aluminium head, water-circulated manifold heater and steel gasket are particularly effective in dissipating the heat and maintaining the temperatures at a moderate level.

The overall cross sectional dimensions of the casting are 5½ in deep × 8½ in wide, except at the front end, where they are 7½ in deep × 11½ in wide to clear the timing drive. Although these dimensions are fairly large, and a smaller and lighter head could have been designed, it was considered to be important to obtain a certain length of port so that the best possible gas flow could be maintained consistently. The head is held down by fourteen studs of En 16T, tightened with torque wrenches set to slip at 54 lb/ft. Their threaded ends are ⅝ in diameter and they are waisted to 5 in diameter, both to reduce stress concentrations at the threads and also to avoid electrolytic corrosion due to contact between the steel and aluminium. The studs fitted to the head have A.N.C. threads at the ends that are screwed into the aluminium and A.N.F. threads at the other ends.

In the early stages of the development, because there are considerable advantages in using as thin a gasket as possible, a ⅛ in thick gasket of an asbestos compound was employed. However, this had several disadvantages, one was that when it was being removed, it tended to break, and pieces truck to the castings and had to be scraped off. Although this trouble can be overcome by the application of a special compound, it was decided to use a cupro-nickel gasket, 0.010 in or 0.015 in thick, with corrugations pressed in it to form the seals. This gasket was entirely satisfactory, for normal operation, but under exceptionally heavy loading experienced in racing and other competition work, it was found that an even stronger one was necessary. Therefore,

a similar type of gasket, but of steel, was developed in conjunction with Coopers Mechanical Joints Ltd. This gasket is about 0.015 in thick and the corrugations pressed in it are deeper than those of the cupro-nickel one. Gaskets of this type have been used successfully in the XK engines with compression ratios up to 12:1.

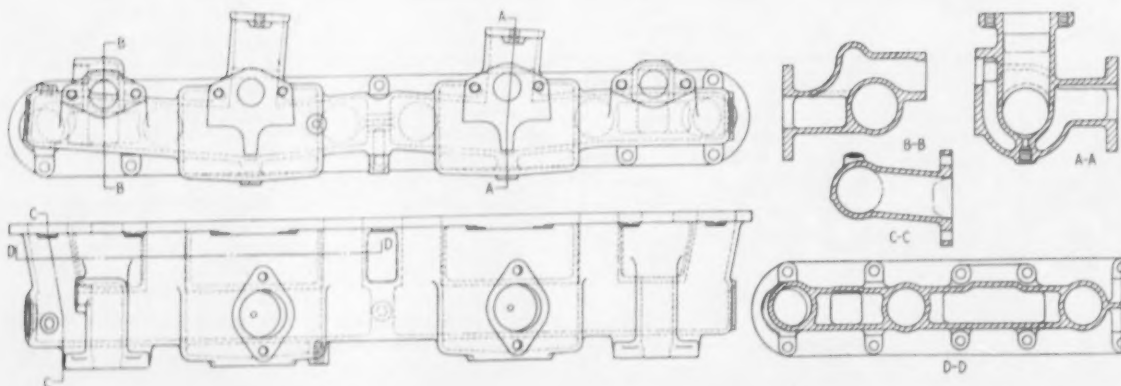
Weslake and Co. Ltd. carried out the design and development of the cylinder head and valve ports. This work was based mainly on flow tests carried out on full scale models of either wood or aluminium. Adjustments to the port shapes and sizes were made until the maximum flow was obtained. This method of obtaining the port shape represents a considerable economy in development time, as compared with the method of repeated bench tests. Moreover, by the application of this technique, it is almost invariably possible to improve the output of any engine by about 10 per cent. A larger valve or higher lift does not necessarily produce a greater flow; also, very small changes in port shape can materially effect the flow characteristics. Therefore, to avoid errors in transferring the shape from the model to the drawing and then from the drawing to the pattern, cores were made accurately from the model and supplied to the pattern makers. As a result, the production ports are in no way inferior to the master port.

From the illustration showing the cross section of the engine, it might at first sight appear that the water jacket space round the exhaust valve guides is inadequate. However, it should also be observed that the guides are surrounded by relatively large masses of aluminium, which has a high thermal conductivity and therefore allows the heat to pass freely to the surrounding jacket areas. The manufacturer states that the most important requirement is to design the jacketing round the exhaust valve seats, guides and ports in such a way that the flow of water is relatively fast. This prevents local overheating due to the formation of vapour in stagnant pockets of water.

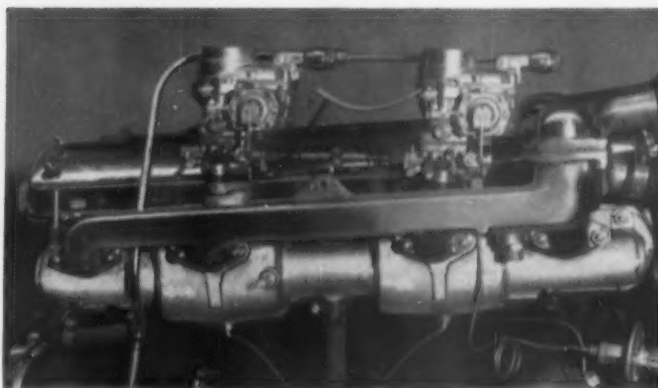
The axes of the exhaust ports, as viewed from above, and the centres of the inlet valve seats, are in transverse planes normal to the longitudinal centre-line of the block and in line with the points of intersection of that centre-line with the cylinder axes. However, the axes of the outer ends of the inlet ports are offset ⅛ in from that plane and their inner ends are swept round to the seats so that the flow into the combustion chamber is directed tangentially relative to the cylinder bores. The sparking plug in each cylinder is directly in the path of this tangential flow. In other words, it is not in line with the axis of the cylinder, but is offset along the longitudinal centre-line of the head. This not only gives good combustion characteristics, but also avoids

* D. DOWNS and J. H. PIGNÉY: "Experimental Investigation into Pre-ignition in the Spark Ignition Engine", Proc. I.Mech.E., Automobile Division, 1950-51.

There are two tracts in the induction manifold, one beneath each riser, and they are arranged in such a way that under cold starting conditions, any fuel that collects in them is immediately drained away



Right: Two Solex B32-PBI-5/S, downdraught, open-choke carburettors are fitted, and they are each equipped with a hand-operated strangler coupled with a fast idle system



Below: A view of the engine installation showing how the intake to the air cleaner and silencer is taken through a hole in the wing valance panel; thence it goes to a small grille at the front of the vehicle



limiting the sizes of the valve throats to accommodate the sparking plug between the seats. Although the cylinder head casting is not given any heat treatment other than for stress relief, it has not been found necessary to use inserts to receive the sparking plugs.

Carburettors, induction and exhaust systems

Two, Solex downdraught, B32-PBI-5/S, open-choke carburettors are fitted. They are equipped with a hand-operated strangler coupled with a fast idle system. A choke diameter of 24 mm has been adopted. Fuel is supplied from the 12 gallon tank by an S.U. electric high-pressure pump. A long, cylindrical type air cleaner and silencer of AC manufacture is employed. It is fitted on an inverted T-shaped duct of rectangular cross section, which fits over the intakes of the two carburettors. The air is drawn in through a duct from a small grille on the front panel of the body. This not only improves volumetric efficiency by taking cold air, but also assists in silencing the intake.

A cast aluminium alloy induction manifold of the straight rake-type is employed and a water rail of the same material is bolted to it. This rail has four flanged ports along its length. Two are the outlets from the head through the water jackets round the junctions between the risers and the induction gallery, while the other two, one at each end, are outlets from the cylinder head through cored ducts over the induction gallery in the manifold.

Both downdraught risers are fairly long and are at right angles to the induction tract, which is of circular cross section. The front riser is between the induction branches

of the pairs of cylinders numbers 2 and 3, and the rear one is between numbers 4 and 5. Thus, each carburettor serves three cylinders. Between the portion of the tract that serves the front three cylinders and that which serves the rear three is a $\frac{1}{2}$ in diameter balance orifice. Because of the transmission angle of the engine, the riser for the rear carburettor is longer than that for the front one, and the main tract is so arranged that, when the engine is installed, the floor of the tract to the rear of each riser is horizontal, while the floor in front of each riser slopes down towards the T-junction. This prevents liquid fuel from collecting at the rear end of each half of the tract under cold starting conditions. A drain tube is provided under each of the risers.

Two three-port cast iron exhaust manifolds are fitted. As viewed from the side of the engine, the branch ports of cylinders numbers 3 and 4 are vertical, while those of numbers 1 and 2 and numbers 5 and 6 are inclined inwards to join them. About 6in from the flanges at the lower ends of the two manifolds, the down pipes join a single pipe, which leads to an elliptical section exhaust silencer. The employment of two separate exhaust manifolds, rather than having all the cylinder head ports discharging into a single manifold, adds to the efficiency of the unit, and there is less likelihood of trouble due to thermal expansion.

Water pump and cooling system

The water pump is of conventional design but, nevertheless, considerable work has been done on it in the development stage, largely to prevent cavitation at high speeds. The pump runs at 0.9 times engine speed, mainly because of the requirements of the 15in diameter, four-bladed, pressed steel fan, the cast iron boss of which is pressed on to the front end of the spindle. A $\frac{1}{2}$ in wide $\times \frac{1}{8}$ in thick V-belt, the included angle of which is 38 deg, drives the D.T.D. 424 pulley spigoted on the fan boss.

The spindle is a plain cylindrical component, $\frac{1}{2}$ in diameter, so that it can be finished by centreless grinding. There are three grooves in it; two, spaced $1\frac{1}{8}$ in apart, form the inner races for the ball bearings and the third locates a rubber thrower ring that operates in a drainage space between the bearing assembly and the water seal. A single tubular component forms the outer races of the two rows of ball bearings; it is sealed at each end to prevent the entry of corrosive or abrasive matter. Location of the assembly is effected by a conical-ended grub-screw, with a lock nut, assembled radially into a tapped hole in the cast iron nose piece of the pump body to register in a hole in the tubular outer race.

A conventional spring-loaded rubber water seal, supplied by the Morgan Crucible Co. Ltd., is employed. It is housed in a pressed brass cup in an aperture in the front wall of the water inlet volute and has a graphite ring bonded to its rear face. This ring bears on the front end of the rotor boss. The rotor is of B.S. 1452 grade 10 annealed cast iron and is 2½ in diameter. It has only four vanes which, to prevent cavitation, are arranged spirally instead of radially. Their inner ends, that is, between the boss and the portion shrouded by the front wall of the impellor housing, are machined away. This feature prevents shock at the entry, where the water flow changes from the axial to the radial direction. Further improvement of the flow characteristics is obtained by streamlining the leading edges of the blades. The pump body forms only the front cover of the rotor housing, the main portion of which is cored in the front face of the aluminium alloy, timing drive cover.

The water flows from the pump outlet into a gallery cored, on the exhaust side, in the upper portion of the cylinder block. From this gallery the water passes vertically through ducts in the block and cylinder head casting; these ducts direct the flow round the exhaust valve seats and guides. Thence it passes transversely across the head, past the inlet valve ports and out through four holes in the side of the head. A thermosyphon circulation through ducts between the inlet side of the head and block cools the cylinder barrels. Two of the outlet holes in the side of the head lead into the jackets round the junction between the induction manifold risers and the main tract, whence the water passes into the outlet rail; the other two, one at each end, lead directly into the water rail. Cast integrally on the front end of the rail is the housing for the thermostat valve. There are, of course, two outlets from this housing, one to the pump inlet and the other to the radiator.

The capacity of the cooling system is 20 pints. A finned tube radiator is employed. Its frontal area is 293 in², and the thickness of the block is 70 mm (2.75 in). An AC valve is fitted to maintain the pressure in the system at 4 lb/in². The thermostat lifts at 70–75 deg C.

Oil pump and lubricating system

A Hobourn-Eaton pump is partly submerged below the oil in the sump. The outer rotor is 2½ in outside diameter and both are 1½ in long. The pump drive spindle, which rotates at half engine speed, is ½ in diameter × 3½ in long; it is secured to the rotor by a ½ in diameter peg. At the upper end, where the spindle projects through the pump casing, it is machined to form a square section and is connected, by a sleeve with a square section hole through it, to the square section lower end of the case hardened En 32B upper portion of the drive spindle.

This upper portion of the spindle has a CP6 or B.S.241 phosphor bronze spiral gear keyed and spigoted on to its lower end and retained by a tab washer and nut on a thread just above the squared section. It is carried in a single 3½ in long × ½ in bore flanged bush, of chilled-cast CP5 phosphor bronze, inserted from below into a boss in the crankcase. A cyanide hardened, En 32A thrust washer is interposed between the lower end of the bush and the gear. Oil is fed through a drilling from the main gallery to an annular groove round the bush and thence through radial holes to the bearing surface. Another annular groove is machined in the bore of the bush mid-way between its ends. This serves as a reservoir to retain the lubricant while the engine is stationary. At the upper end of the spindle the diameter is increased to 1 in and it is counterbored and slotted for the spigot and tongues of the distributor drive.

From the pump, the oil is delivered to a connection under the web that carries the centre main journal bearing. Thence it goes through the felt element of the Tecalemit full flow filter, mounted on the side of the cylinder block. The relief

valve, which is incorporated in the filter, lifts at 50 lb/in². From the filter, the oil under pressure goes to the ½ in diameter main gallery on the right-hand side of the engine.

All the oil passages are of large diameter to ensure that even under the coldest conditions none of the bearings is starved of oil regardless of its distance from the pump. Wherever possible, the ends of the holes are countersunk. This is to remove the sharp brittle edges of the holes in the iron castings so that they cannot be broken away by the oil flow and damage the bearings. Each main bearing is served by a ½ in diameter hole, from the gallery, through which the oil passes to a groove round the bearing housing. This groove is machined eccentrically relative to the axis of the bearing to clear the bearing cap bolts, which are offset to one side so that it is impossible to assemble them incorrectly.

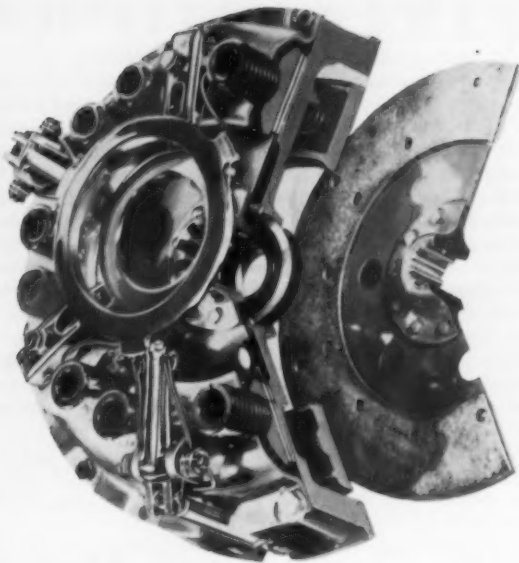
Four holes are drilled radially through each two-piece bearing shell, and the oil goes through these into a shallow groove on the bearing surface. From this groove, the oil passes not only to the main bearing surface, but also through a ½ in diameter hole drilled through the crankshaft to the adjacent crank pin. Because of the relatively heavy loading on the centre main bearing the drillings to the crank pins are taken only from the intermediate, front and rear journals.

As can be seen from the illustration showing the longitudinal section of the engine, the holes in the crankshaft are arranged in such a way that a ½ in diameter centrifuge sludge-trap is formed in the crank pin. Also, the feed to each big end bearing is through a ½ in diameter hole drilled at right angles to the plane containing the major axes of the adjacent pair of webs. Experimental work has shown that at high speeds the oil flow through the big ends is considerably reduced by drilling the journals in this manner in preference to drilling straight through from the main bearings to the outer faces of the crank pins. Excessive amounts of oil would otherwise be flung out of the bearing by centrifugal force, not only because the clearance at the outer face of the bearing is larger than at the sides, where the oil holes break out of the journal, but also because there is a tendency for the oil in the vertical drilling through the connecting rod to be flung into the small end bearing.

Each camshaft is served by a ½ in outside diameter × 20 S.W.G. pipe, which delivers the oil in through a hole drilled longitudinally from the rear face of the casting to join a vertical duct to the rear bearing. Thence the lubricant goes into the hollow shaft and through radial holes to the other bearings. An oil bath is formed in the tappet chamber on each side to receive the overflow from these bearings and to provide for the lubrication of the sliding faces of the cams and tappets. The tappet guides stand proud of the base of each chamber so that they are not flooded with oil, and the oil level is regulated by three ducts, one between each pair of valves, that form weirs over which the oil drains to the cell that houses the valve springs. Finally, the oil passes to the ends of the head and down through ducts to the timing case and sump.

Some of the oil draining from the front end of the head is splashed into a trough between the two arms on top of the spider, to lubricate the eccentric jockey sprocket. Oil from this trough passes through a vertical hole into an annular groove round the front end of the spindle of the eccentric. Then it goes through a radial hole into an axial one and thence through three diametrically drilled holes to the bearing.

The intermediate sprocket is lubricated in a similar way, but in this instance there are two holes to receive the oil splashed down from above. One is drilled from the top face of the front end of the block to a space between the block and the rear end of the spindle; from this space, the oil passes directly into the axial duct in the spindle. The other hole is drilled from the top of the front cover; it passes oil into an annular groove round the front end of the spindle and then through a radial hole to the axial one.



Borg and Beck combined power take-off and main drive clutches sectioned to show the general arrangement and main components

Dual Purpose Clutch

the normal way. The unit nearest to the flywheel is used to transmit the main drive to the rear wheels, while the other is for the power take-off. There are two main advantages to this arrangement: one is compactness and the other is that both clutches are in a favourable position to transmit the drive.

Both clutches are more or less conventional, so far as principle of operation is concerned. They are each actuated by an independent fork and a ball thrust bearing. The fork for the transmission clutch is actuated in the conventional manner by the clutch pedal, and the power take-off control is a hand lever to the near-side of the driver. Power is transmitted from the main drive clutch by a solid splined shaft to the primary shaft in the gearbox, while the power take-off clutch is connected by means of a hollow splined shaft to flanged dog-couplings and thence to a hollow pinion that drives a gear on an intermediate, power take-off shaft.

Power take-off clutch

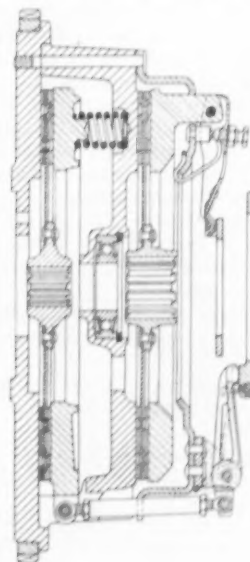
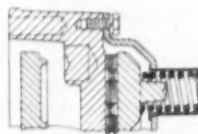
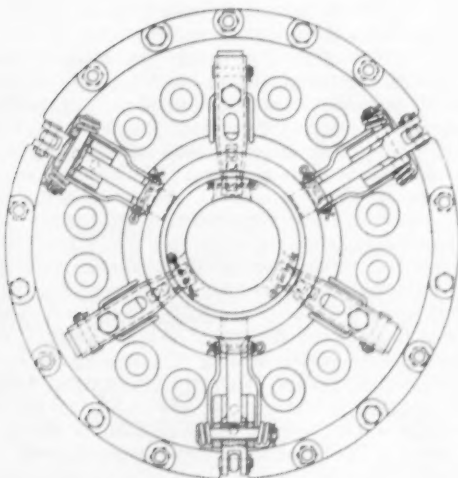
The principal components of the power take-off clutch are: a pressed steel cover, a pressure plate, twelve thrust springs carried in cups, and a driven plate. This driven plate carries a splined hub, to which a disc carrying the two friction facings is riveted. The power take-off cover is attached to the main drive cover by six set screws and spring washers.

Three lugs on the pressure plate extend through slots in the cover. These lugs transmit the drive from the cover to the pressure plate and also carry the pivots for the release levers. A short distance inboard of the pivot at the outer end of each release lever there is an adjustment screw and lock nut, while a leaf type retainer spring is riveted to the inner end of the lever. The forked lugs of the release plate

A Unit Comprising Two Clutches in Tandem, one for the Main Drive and the other for Power Take-off

A NEW Borg and Beck 11in diameter dual purpose clutch, manufactured by Automotive Products Ltd., of Leamington Spa, is now fitted to the Universal Tractor made by the Nuffield Organisation. This double clutch, which in effect is two units mounted in tandem, is bolted to the flywheel in

In the new Borg and Beck unit, the power take-off and main drive clutches are mounted in tandem on the rear face of the engine flywheel



fit over these inner ends and are retained by split pins passed through holes in the lugs in such a manner that the springs bear against the pins and force the ends of the levers into positive contact with the plate between the lugs. There are also three wire springs, one on each lever; they are passed through holes in the levers, at points adjacent to the adjustment screws, and their ends are hooked under the edge of the hole punched in the centre of the housing. The function of these springs is to ensure that the adjustment screws are always in contact with their seats, which are riveted to the casing to act as the fulcrums for the levers.

Main drive clutch

A sturdy cover encloses the main drive clutch. Housed in the centre of the rear wall of this cover is a grease-packed ball bearing that carries the end of the power take-off shaft. The assembly inside the cover comprises a pressure plate, twelve thrust springs seating on heat insulation washers, and

a driven plate. This plate is riveted to its splined hub, and has two friction facings.

Three turnbuckles connect the pressure plate to the outer ends of the release levers. Each of these levers is bushed to carry a headed pivot pin and is mounted in a yoke riveted to the rear face of the cover over the power take-off clutch. The connection between the release levers and the turnbuckles is effected by left-hand threaded eyebolts, screwed into the ends of the turnbuckles, and headed pins. Wire springs retain the inner tips of the release levers against their plate. These springs are secured by split pins to the inner ends of the levers and at the outer ends are wrapped round the ends of the pivot pins and bear against the yokes. Right-hand threaded eyebolts are screwed into the other ends of the turnbuckles, the connection to the pressure plate being made by headed pins passed through holes in the forked lugs on the periphery of the plate. These lugs project through slots in the cover to furnish the drive.

International Conference on Fatigue of Metals

A Conference at Which 70 Papers, All by Leading Experts on Fatigue, Will Be Presented

AN International Conference on Fatigue of Metals is being sponsored by the Institution of Mechanical Engineers in co-operation with the American Society of Mechanical Engineers. Meetings are to be held in London from the 10th to the 14th September, 1956, and in New York from the 28th to the 30th November, 1956. In London, the meetings are from 10 a.m. to 12.30 p.m. and 2.30 to 5 p.m.

It is also planned to arrange visits to centres at which research on fatigue of metals and metal components is being conducted. Persons registered to attend the Conference may take part in any number of these visits. Expenses will have to be paid by the registrants but the Institution will arrange transport and accommodation if required. It is expected that there will be about eight visits to places in England and Scotland from the 17th to the 21st September inclusive.

Those wishing to register for attendance at any or all of the technical sessions will be required to pay a registration fee of one guinea. An additional fee of one guinea will be payable by those wishing to take part in the visits. These fees are intended to cover administration costs. Advance copies of the Papers will be available by August 1st. They will cost £1 for the complete set and 3s. 6d. for the Papers of any one session. Bound volumes of the Proceedings will be on sale later. These will contain reports of the opening and closing sessions, the papers and discussions, and the Reporters' introductory notes, together with name and subject indexes.

With regard to registration, preliminary reply forms will be issued in May with *The Chartered Mechanical Engineers*. On receipt of these, the Institution will forward to applicants a detailed reply form, to be returned with the remittances. All registered members will receive a programme of the Conference.

The opening and closing sessions of the Conference will be held at the Central Hall, Westminster, and the other sessions will be in the Meeting Hall of the Institution of Mechanical Engineers. Owing to the large number of papers to be presented, it will not be possible for authors to

give them in person. However, it is hoped that there will be sufficient time for the authors to make brief replies to the discussion. The papers will be presented in abstract by Reporters. About 30 minutes will be allowed for reporting in each session, so there will be about two hours for discussion and replies. The introductory address, on the other hand, will be delivered in person by Dr. H. J. Gough. Lists of speakers will be prepared before the meetings and any registered member may ask to be placed on the list. Speakers in discussions will be furnished with proofs of their comments before the Proceedings are compiled. At the follow-up Conference in New York, the Papers will be presented again for discussion.

Out of a total of 70 Papers, 40 are from Great Britain and Commonwealth, 20 from the United States and 10 from other countries. Session 1 on Monday morning, the 10th September, is the opening of the Conference by the President, Mr. T. A. Crowe, M.Sc., an outline of the arrangements, and an introductory address by Dr. H. J. Gough, C.B., M.B.E., F.R.S. (Past President). On Monday afternoon and Tuesday morning, sessions 2 and 3 on "Stress Distribution" will be held. These are followed on Tuesday afternoon by session 4 entitled "Temperature, Frequency and Environment". The following day, the morning session is on "Metallurgical Aspects of Fatigue" and the afternoon session is devoted to "Basic Studies". Session 7, on the morning of Thursday, the 13th September, is likely to be of special interest to automobile engineers, since it is entitled, "General Service, Automobiles and Specific Components." Session 8 in the afternoon is on "Air Frames and Engines," and session 9, on the following morning, deals with "Railways, Marine Engines and Welding." These last three sessions come under the general heading "Engineering and Industrial Significance of Fatigue." On the afternoon of Friday the 13th September, session 10 will comprise reviews by Reporters and a summary of the problems that have yet to be solved. The summing-up, to which 2½ hours will be devoted, will enable each reporter to speak for 15 minutes. Then the Conference will be closed.

WHITELEY POWER-ASSISTED STEERING

A Development with the Valves Incorporated in the Piston of the Jack

MENTION of the Whiteley power-assisted steering unit was made in the 1955 London Show Review issue of *Automobile Engineer*. Since then, further details have become available, and the general arrangement of the latest version of the unit is shown in the accompanying illustrations. This power-assisted steering system is based on a hydraulic jack. The end of the cylinder of the jack is attached by a ball joint to the vehicle structure, and the outer end of the tubular piston rod is screwed into a forging that carries a ball joint connection to the steering drag rod. Inside the piston rod is the valve rod, the outer end of which is connected by a ball joint to the steering drop arm. This ball joint is housed in the hollow forging on the end of the piston rod, but its movement, and therefore that of the valve rod, relative to the forging, is limited by stops, so that direct manual operation can be obtained in the event of failure of the hydraulic system. Direct operation is also required, of course, when the engine is not running, for example, if the vehicle is on tow.

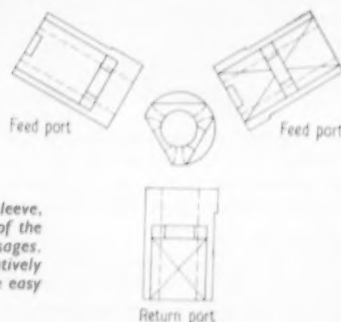
Two hydraulic connections, one for the supply and the other for the return, are made by two flexible pipes to the forging that carries the drop arm and drag rod ball joint assemblies. Each of the two unions communicates with a separate annular groove in the periphery of a sleeve that is fixed in this forging. Pressure oil passes from the groove in the periphery, through a radial hole into an annular groove in the bore of the sleeve and thence through a radial hole into the hollow valve rod. The ends of the annular space in the bore of the sleeve are closed by U-section oil seals. Since the range of movement of the rod is small, the radial hole through which the oil passes can in no circumstances move past the oil seals and thus out of communication with the annular space. Because the sleeve does not move in the forging that carries the ball joint connections to the drop arm and drag rod, simple circular-section rubber rings in grooves on each side of the main oil passage are adequate to seal its outer periphery.

Returning oil passes from the annular space between the piston rod and valve rod, into the bore of the sleeve and annular groove round the periphery of the sleeve and thence to the return pipe. Since this is the low pressure side of the system, circular-section rubber rings are employed as seals between the end of the piston rod and a shoulder in the bore of the sleeve and also between the end of the sleeve and a shoulder in the forging that carries the assembly.

At the other end of the rod assembly, the piston is screwed on to its rod. It is hollow and houses the valve sleeve, in which the slide itself moves. There are three annular grooves round the slide. Radial holes are drilled from the

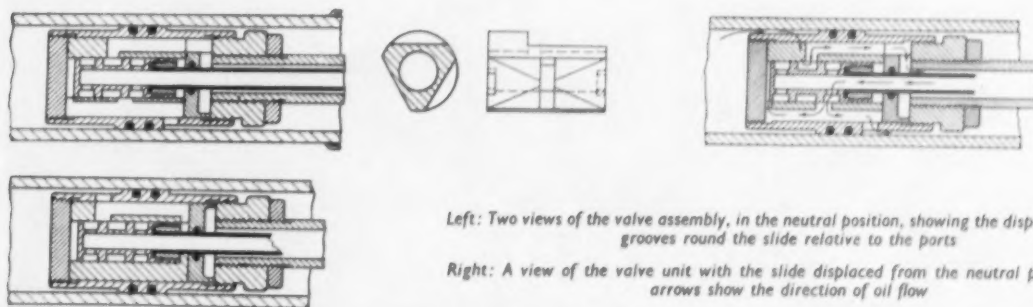
bases of the two outer grooves to communicate with an axial hole in the slide. The valve rod is secured in the end of the slide in such a way that high pressure oil passing down the centre of the rod flows directly into the axial hole in the slide and thence through the radial holes to the two outer grooves.

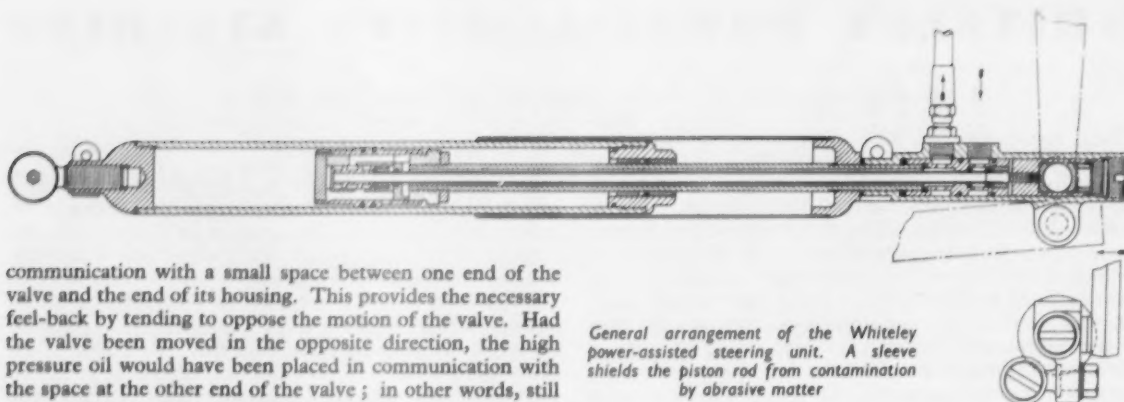
The central groove simply acts as a transfer passage to connect the port from one side or the other of the piston to the return system. There are three flats, 120 degrees apart, round the periphery of the valve sleeve, and the return oil passes from the cylinder along one of two of these flats, according to whether the slide has opened the port serving one side of the piston or the other, and thence through the



open port, round the central groove in the slide valve and out through another port to the third flat. Thence the oil passes along this flat and into a space between the end of the slide valve housing and the end of the piston rod. From this space it flows between the inner and outer rods, for the piston and the valve respectively, to the other end of the unit and out through the flexible return pipe.

In the neutral position, the valve is centrally disposed between the ends of the sleeve; at the same time, the lands between the pressure grooves and the central, return groove are directly in line with the ports that communicate with each end of the piston. Since these ports are larger than the lands, oil can flow directly from the outer grooves into the return groove, and thence along the third flat into the annular space between the piston rod and valve rod. If the valve is displaced from the neutral position, as in the illustration below, one of the high pressure grooves is placed in direct communication with the port to one side of the piston, the other is blanked off, and the return groove is in such a position that the oil from the other side of the piston can flow through it to the passage communicating with the return system. The high pressure oil is also placed in





communication with a small space between one end of the valve and the end of its housing. This provides the necessary feel-back by tending to oppose the motion of the valve. Had the valve been moved in the opposite direction, the high pressure oil would have been placed in communication with the space at the other end of the valve; in other words, still in opposition to the motion of the valve.

The main advantages claimed for the system are as follows. In the neutral position, there is no direct mechanical connection between the road wheels and the drop arm of the steering box. However, as soon as the road wheels are deflected a small amount due, for example, to their hitting a bump in the road, hydraulic pressure is immediately built up to resist the motion. This tends to damp out shocks that would otherwise be transmitted directly to the steering wheel. Another good feature is that there can be no tendency for the power jack to take charge and to continue to move the steering system after the driver has stopped turning the

General arrangement of the Whiteley power-assisted steering unit. A sleeve shields the piston rod from contamination by abrasive matter

steering wheel. This is because any continuation of the motion of the jack would cause a reversal of the valve position, and therefore automatic correction. Since there are no springs in the valve unit, it is inherently free from vibration troubles. The piston rod is chromium plated and is protected from abrasive matter by a sleeve, mounted on the forging that carries the ball joints for the drop arm and drag link and extending over the end of the jack cylinder. Thus, the rate of wear of the piston rod, and the risk of damage to the oil seal by abrasive particles are negligible.

ALL-ALUMINIUM LORRY BODY

THE illustration below shows a special purpose body, built for T.I. Aluminium Limited entirely from "Timinium" alloys and fitted to a Leyland Octopus chassis. The lorry's duties are governed by the need for dealing with the Company's present expansion of business. Thus the design of the body had to make provision for such different loads as timber cases of aluminium sheet; foundry billets, up to 14 tons load; and scrap and swarf often containing heavy items.

The body is 24ft 6in long by 8ft wide and 3ft deep. Timinium alloy is used for the cross-bearers and the heavy-duty flooring is bolted directly on to them. Three 8ft sections form the sideboards. Each section consists of two

panels hinged together. The top panel can be dropped and secured to the lower one and the two together lowered as one 18in unit.

The scrap and swarf load is segregated, prior to loading, into three classes. Each class must be kept apart, and two removable partitions of heavy-duty corrugated section accordingly divide the body into three 8ft compartments. Each partition slides down into place in channel guides, built into the intermediate pillars that retain the sideboards when they are erected. These four intermediate pillars, and the four corner pillars, are themselves detachable. The weight of the complete body is only 37 cwt, a very low figure for a payload of 14 tons.



Pneumatic Gauging

*Diverse Applications of
the Mercer Dial-indicating,
Back-pressure
Type Instrument*



Fig. 2. Standard and miniature single-dial instruments have 6 in diameter and 4 in diameter dials respectively

ALTHOUGH pneumatic gauging has only a relatively short history—the patent literature referring to dimensional gauging by means of air or liquid pressure extends back less than 40 years—its present application is widespread. Numerous characteristic features contribute to the general adoption of the system in its several methods. It is uncommonly versatile in application and functions equally well on large or small work and for either internal or external dimensions. A single set of equipment can be used for a variety of different gauging operations by the provision of different gauging heads which are attached by means of a simple plug-in connector. Response is rapid and the indicating means, whether a dial or a column, can be arranged some distance from the gauging point. The height or depth of the gauging point in relation to the instrument has no influence in a high-pressure system, and the normal variations of shop temperature will not affect the accuracy of the reading. The lag is less than 1 sec for instrument-mounted gauges or immediately adjacent gauging fixtures, and rather more for a gauging head on a 4 ft flexible lead. Operation up to 15 ft is practical, though with a slower response, of course. In any case indication is exceptionally clear on relatively wide scales and does not call for close reading. Where multiple gauging operations are performed, indicating instruments are commonly grouped for rapid scanning. The magnification can be varied over a wide range in order to provide an open scale or a wide spacing of tolerance points.

Despite the fact that an air gauge is suitable for measuring fine work in metrological laboratories, and for repetitive measurement of components to 0.00002 in on such items as parts of fuel injectors for diesel engines, it remains essentially simple as regards its pneumatic circuit and also in the construction of its relatively few component units. In theory there is no direct contact between gauge and work, though it may not be possible to avoid some guiding contact on non-vital areas of the gauge. As a consequence it is possible to gauge over highly polished or relatively soft surfaces without damage and the gauging elements are assured of long life. The flow of air keeps clear the conduits and nozzles in the gauge head and also tends to remove foreign matter from the immediately adjacent surfaces of the work.

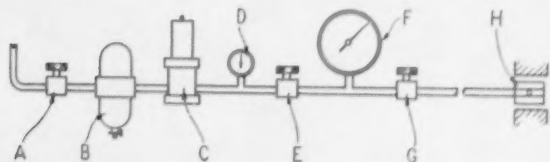
The foregoing features, together with the fact that no special skill is required of the operative, make it particularly suitable for use in the workshop, either on the bench or directly on the machine, for checking work in progress. It is also readily adaptable for the automatic gauging of work in continuous production. On some fully mechanized transfer lines automatic air gauging is installed between

Fig. 3. Plug gauge for gudgeon pin bore in piston. Fitted to standard adaptor on instrument



Fig. 1. Diagram of air circuit

A Stop cock B Air cleaner C Pressure regulator D Check gauge E Control orifice
F Pressure gauge, calibrated in equivalent linear units G Zero control H Gauging head



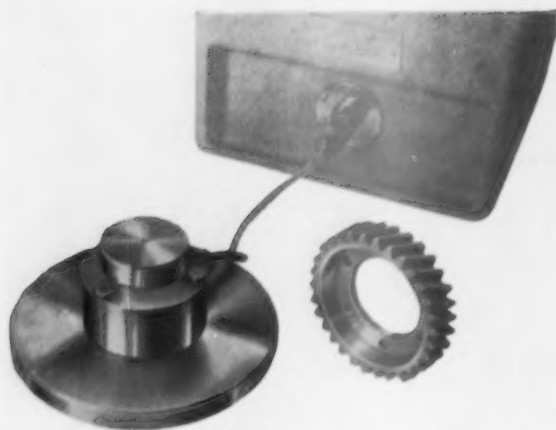


Fig. 4. Bench-type plug gauge for the shallow bore of a gear wheel

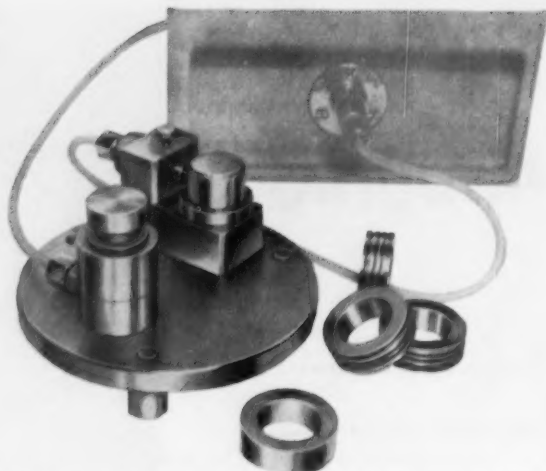


Fig. 5. Gauging unit for checking squareness and concentricity of the bore of an annular part

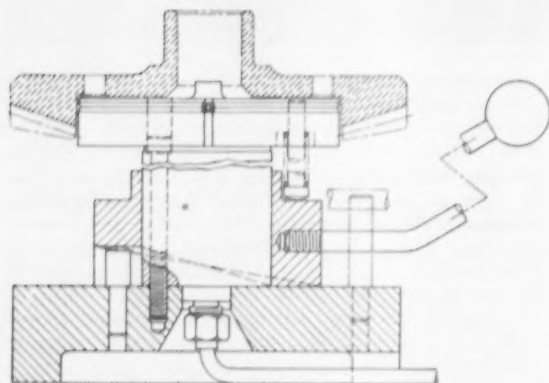


Fig. 6. Special fixture for the rapid checking of a shallow recessed bore in a crown wheel

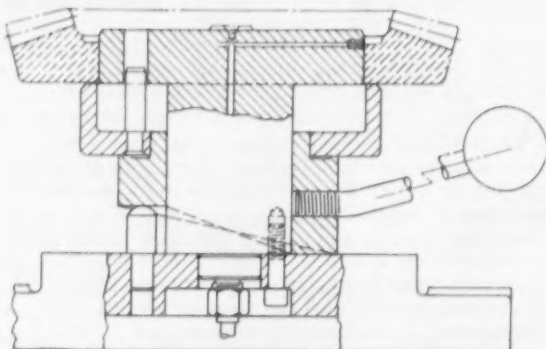


Fig. 7. Lowering and elevating fixture for gauging crown wheel bore

successive operations to check for conformance to tolerance, to warn, either audibly or visibly, when limits are approached as a consequence of tool wear, and to stop the machine in the event of faulty work being produced as a result of loss of adjustment or tool breakage.

Air gauging was introduced to meet the needs arising from mass production. Its adoption received a marked impetus during World War II and subsequently the development of high-rate flow production methods and the demand for still finer limits of tolerance have led to its more common use in the production line and also in multi-unit gauging stations to group products ready for selective assembly. For use on, or in conjunction with, transfer machines it has tended to become an essential component and in any approach to the goal of automation it would seem to be indispensable. Air gauging may be said to be able to meet many current requirements and also, it is anticipated, will provide a solution to the problems of automatic control in the immediately foreseeable future.

Various air systems

A number of different air circuits for pneumatic gauging have been developed and several have attained commercial application. All take compressed air from a shop line, pass it through a regulating valve and deliver it at a constant value to one or more metering orifices in the gauging head. The escape of air from these orifices is influenced by the proximity of the work being gauged and the effect, in the

form of variation of back pressure or of flow rate, is indicated by appropriate equipment on a suitably calibrated scale.

Apart from variants and combinations, most systems will be found in one of two main groups; the original back-pressure type, most common in Britain and in Europe and the flow type which is widely used in America. The back-pressure systems may conveniently be divided into two sub-types; the high-pressure system indicating on a Bourdon-tube pattern dial gauge and the low-pressure system in which pressure variation is indicated by a water column.

Irrespective of type or air circuit, an air gauge is not a measuring instrument but fundamentally is a comparator. The instrument is calibrated against masters and indicates plus and minus deviations in the work from the dimensions of the masters. This characteristic is actually of great advantage since the geometrical features of the work and also of the wearable surfaces of the gauging head affect the value of the air discharge coefficient and, consequently, the gauge indication. The use of masters also permits the replacement of gauging elements, or even the substitution of completely different gauging heads, without the necessity of re-graduating the scale of the instrument.

The Mercer air gauge

Mercer instruments are of the back-pressure type with dial indication. Air is supplied from the shop line at a pressure of from 60 lb/in² to 120 lb/in² and through a special air purifier to a precision, diaphragm type pressure

regulator. From the downstream side of this unit the air at a constant pressure of 40 lb/in² passes to a check pressure gauge, and to a variable control orifice to permit flow regulation to suit different gauging heads within the range of the instrument. Thereafter, the air stream passes to a sensitive pressure gauge, a variable bleed to atmosphere (for zeroing the instrument) and finally, by way of a flexible conduit, to the gauging head. Components and controls are housed in a cast light alloy casing having a commendably smooth exterior. The indicating dial is arranged in a plane inclined to the vertical for convenient reading and the conduit to the gauging element is attached by a plug-in adaptor at the front of the instrument and secured by a bayonet latch. A sunk carrying handle is provided at the top of the casing. Overall dimensions of the standard instrument are height 11½ in, width 9 in and depth 10½ in. The weight is approximately 30 lb.

An instrument of reduced size, shown alongside a standard unit in Fig. 2, is available for direct installation on machine tools, for wall mounting, for use on inspection benches and other locations where close reading is possible or where space is at a premium. It has a 4 in diameter dial, instead of the normal 6 in diameter dial, and a different layout of the controls, but otherwise resembles the standard instrument. A twin dial unit is also in regular production for simultaneously checking two dimensions on such work as two-diameter bores, spaced diameters on tapers, or the extremities of blind bores. This instrument is 16½ in wide and weighs about 44 lb. Grouped instruments for multi-dimensional checking are built up either into a complete gauging unit or as part of a gauging station, to meet specific requirements. Such assemblies can be arranged to link with coloured light systems or with other visual or audible warning of out-of-tolerance or reject components.

Standard dials have a 270 deg scale and are of the balanced type with a centre zero. They are available for ranges of 0.002 in and 0.004 in, both reading in increments of 0.0001 in and for a range of 0.050 mm reading in 0.001 mm. Dials can, of course, be specially scaled to meet specific requirements. The instrument may be graduated with the zero

disposed to either the plus or minus side, with coloured bands or segments, or other variations peculiar to mass production systems or where unskilled labour is used. A special 180 deg high magnification scale is graduated for direct reading to 0.00005 in.

Pneumatic gauging elements

The actual gauging elements carrying the metering nozzles are designed to suit the component to be gauged and, consequently, may be of widely differing size and shape and may be arranged for either manual or mechanical application. Despite this variety, however, elements can be classified into three general types. In the first, an air jet is applied directly to the work which is not in contact with the gauging head at the point of measurement. Gauging heads in the second group embody a mechanical component to contact the work at the measurement point and indirectly influence the air jet. The third group is more limited in its application and is generally used to determine the cross-sectional area of a bore or an orifice rather than its dimensions or shape. The bore of the work constitutes the air escape area and should this area be beyond the capacity of the instrument it may be reduced by means of a suitable imperforate restrictor.

Plug Gauges

The principle of the direct air gauge is that the air flow, and consequently the back-pressure, is approximately proportional to the escape area. This annular area is determined by the circumference of the nozzle bore multiplied by the clearance between the nozzle end face and the work. In multi-nozzle gauging heads it is essential that nozzle dimensions are balanced in order that displacement of the work will result in only a negligible change of indicated values. It will be appreciated that as the work is displaced from one nozzle the escape area is almost exactly compensated by the closer approach of the work to a diametrically opposite nozzle.

Pneumatic plug gauges, probably the simplest and the most widely used of this group of gauges, are frequently designated

Fig. 8. Gauging cylinder bores on the machine table with a hand type plug gauge



Fig. 9. Self-locating plug gauge for splined bores

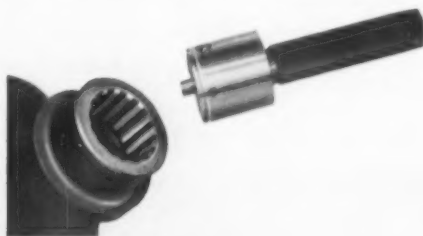


Fig. 10. Two-jet plug for checking the root diameter of splined bores





Fig. 11. Plug gauge for simultaneously checking two diameters of a taper bore

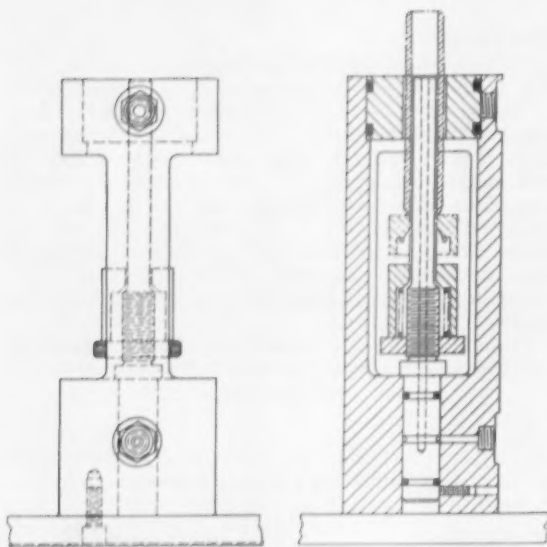


Fig. 12. Gauging fixture for the bore and outside diameter of a valve guide

Fig. 13. Bench type, direct-jet gauge for thin-walled cylindrical parts



"gauging mandrels" or "gauging spindles". Fig. 3 shows such a gauge, mounted directly on the instrument and with its diametrically opposite nozzles lying in a horizontal plane, used for checking the diameter of the gudgeon bore in a diesel engine piston. Manufactured to adequate standards of precision, the mandrel ensures that the axes of the gauge and the work are approximately coincident and that the jets are truly radial in a plane normal to the axis.

When required for checking a true diameter, two diametrically opposite nozzles are used, but three or more nozzles may be provided to obtain average values of the diameter. Special mandrels are available for determining other factors in a bore, such as straightness, lobing or bell-mouth, and gauging fixtures are produced to specific requirements for checking squareness of a face with the axis of a bore, bowing of a face, or concentricity of a bore with an external diameter. Fig. 4 shows typical plug-gauging units for the shallow bore of a helical gear wheel, while Fig. 5 shows a specially built unit for checking the squareness and concentricity of an annular component.

Special fixtures for the rapid gauging by unskilled labour of a shallow recess and a bore in transmission crown wheels are shown in the drawings Figs. 6 and 7. On each of these the component is merely laid horizontally on supporting elements, approximately centred, and then lowered to traverse the bore over the gauge plug by partially rotating a helical face cam member bearing on the cylindrical support column of the gauge. After gauging, the return of the cam member to its original position raises the component clear of the gauge plug to facilitate unloading.

For checking work in continuous production, or for gauging between operations, it is generally more economical to take the instrument to the place of work rather than to bring the work to an inspection bench. A typical example is illustrated in Fig. 8. There an air gauge set up alongside the machine is used for the immediate checking of the bores of a four-cylinder engine block. The diameter of these bores is held to limits of $+0.0004$ in and -0.0006 in.

Splined bores cannot, of course, be checked by means of conventional plug gauges. Special gauging heads, however, are manufactured for the purpose. For measuring the bore, the plug is provided with two diametrically opposite ball locators which are entered into spaces between the splines and thus bring the two air nozzles, arranged at 90 deg to the balls, into correct radial position over the crests of splines to check the bore diameter. A mandrel of different pattern is required for gauging the root diameter. This is formed with, say, six splines which guide and centre it in the component. The air jets are mounted, diametrically opposite, in two of these splines. Gauges for checking the width of individual splines and the width of the spaces between the splines can be made for dimensions down to a minimum of 0.125 in.

A twin-dial instrument is used for the gauging of taper bores. The special gauging mandrel has axially spaced pairs of air nozzles, each on a separate air circuit. Thus, when the mandrel is inserted in the bore the two diameters are simultaneously indicated on the instrument dials. The illustration, Fig. 11, shows the set-up for routine checking of the bores of taper drill sockets.

Plug gauges can be manufactured to check all reasonable bore diameters down to a minimum of 0.0937 in. Every gauge is supplied with two master setting rings to enable the instrument to be set up or checked over the working range of the specific dimensional limits. These rings are nominally made to the upper and lower limits of the part to be gauged. The actual sizes of the rings, owing to manufacturing tolerances, will be within a few hundred-thousandths of an inch of the designated limits. Each ring is precisely measured and its actual dimension is engraved on its side.

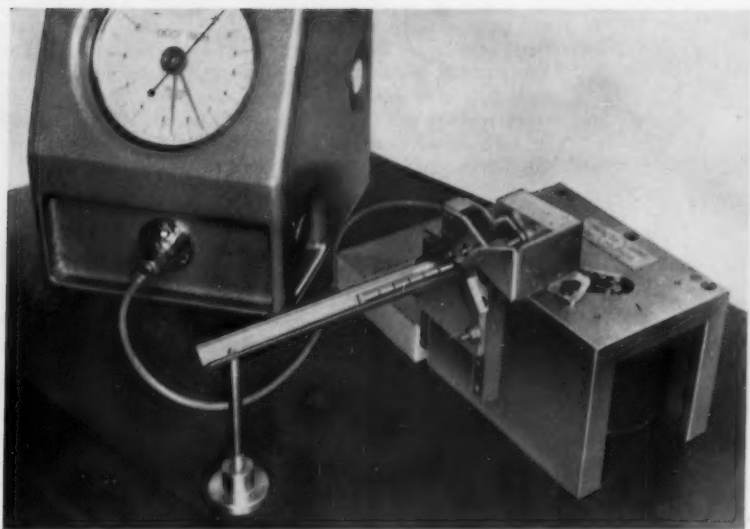


Fig. 14. Mechanized fixture for batch-gauging rollers of fourteen different diameters



Fig. 15. Direct-jet fixture for the continuous gauging of precision strip material



Fig. 16. Mechanical-contact type gap gauge



Fig. 17. Mechanical-contact hand gauge for use in the workshop

Ring gauges

The converse of the plug gauge is the ring gauge for checking the outside diameter of cylindrical parts. Like the plug gauges, they are provided with two diametrically opposed air nozzles for measuring a true diameter or with three or more nozzles to determine an average diameter or lobing. Usually such gauges are mounted on a support having a substantial base and the work is introduced horizontally. Each ring gauge is supplied with two master setting plugs for checking the adjustment of the instrument.

Ring and plug gauges are sometimes used in combination fixtures for the simultaneous checking of external and internal diameters. Such a fixture for measuring the entire length of a valve guide is shown in Fig. 12. In what may be termed a lantern body member, the air mandrel for the bore is mounted in the base and the ring gauge for the external diameter in the head. Slidable over the mandrel is a stop member supported by a helical spring from a screwed ring adjustable for height on the stem of the mandrel. The operator feeds the component through the ring gauge and over the mandrel, depressing the stop against the spring until the full length of the component is within the ring gauge.

On release, the spring raises the component about one inch so that the end protrudes to facilitate unloading.

Direct-jet gap gauges

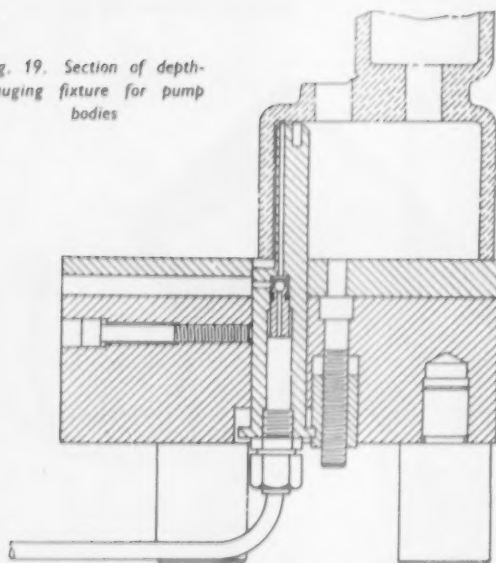
Fixed gap, caliper gauges are produced in a variety of patterns for hand or bench use and also as measuring components for more elaborate gauging fixtures. The jaws are usually faced with hard metal and a back stop is provided to facilitate location on the work. Hand gauges can be used for work in progress on machine tools while the mounted bench gauges are specially useful for the inspection of thin-walled or other relatively delicate parts without the risk of distortion.

Direct-jet gauging fixtures for the external measurement of parts are particularly suitable for the rapid routine inspection by unskilled labour of parts in mass or continuous production and can be developed for automatic gauging systems. Opposed air nozzles are mounted in relation to a chute, track or conveyor belt on which the work is passed through either manually or mechanically. An example is shown in Fig. 14 of a mechanized fixture for gauging the diameter of cylindrical rollers used in fuel injection equip-



Fig. 18. Fixture for gauging the depth of gear pump bodies

Fig. 19. Section of depth-gauging fixture for pump bodies



ment. The rollers are loaded into a chute and slide by gravity to a feed roller faced with a resilient material. This roller is driven by a fractional h.p. motor mounted on the fixture and both its pressure and the angle in relation to axis of the work in the chute can be adjusted to vary the rate at which the work is fed through the gauging position. Rollers of fourteen different sizes ranging from 4.5 to 10.0 mm diameter are gauged with this equipment. The work is batched and the fourteen direct-jet gap gauges to accommodate the different components are interchangeable on the fixture.

An interesting application of the direct-jet method occurs in the continuous gauging of precision strip material. The special equipment for this purpose, developed in conjunction with Glacier Metal Co. Ltd. of Alperton and shown in Fig. 15, can check strip up to 8 in in width and down to 0.004 in in thickness. To facilitate the perpendicular setting of the heads, the yoke is pivotally mounted on the base structure and is stayed in its adjusted position to obviate any possible vibratory movement. From the pivot point

the air lead from the instrument is branched and separate flexible leads are taken to upper and lower air nozzles. The lower nozzle is fixed in the yoke arm but the upper one is mounted in a micrometer head to allow adjustment to accommodate strip of different thicknesses. Strip material to be checked is reeled between the gauging heads under appropriate tension to ensure flatness and straightness.

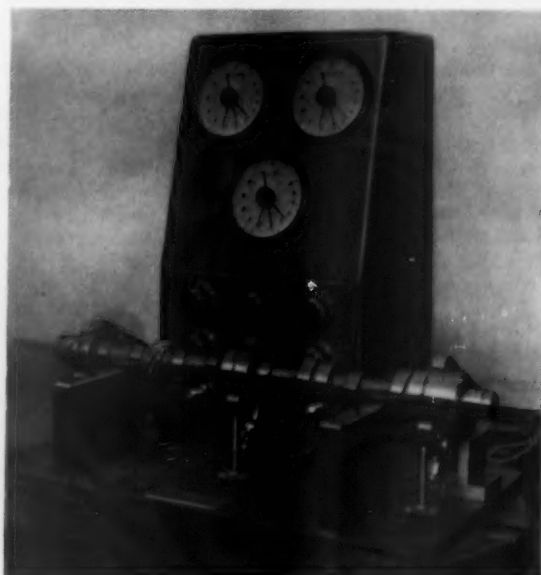
Mechanical-contact air gauges

In many instances mechanical-contact gauges, in which the relative movement of opposed jaw surfaces is measured pneumatically, can offer advantages over the direct-jet type of gauge. They occupy a minimum of space, can be adapted to provide special jaw forms for awkwardly located or closely confined work, and can measure close up to the corner of a shouldered workpiece. Since such gauges are readily adjustable over a range of ± 0.015 in from the mean dimensional capacity, wear on the jaws can easily be accommodated. It is common practice, however, to face the jaws with hard steel, hard chrome, or tungsten carbide. The typical gauge in Fig. 16 has a fixed lower jaw and a movable upper jaw.

Plug gauges can also be of the mechanical-contact type. In this form they are specially suitable for workshop use, routine inspection by unskilled labour, or automatic gauging in the machine line.

Mechanical-contact gauging devices are made for a wide variety of special applications, such as the depth of blind bores or irregular orifices. Of considerable interest is the fixture for gauging the critical depth of a gear pump body. The component is placed on the striated table and tracked over the upstanding gauging head, so that the end wall closing the bore is traversed to check flatness. One side of the head is machined away so that the contacting end of the plunger can closely approach the corner of the bore. In the illustration, Fig. 18, the fixture is shown connected to the gauge instrument and alongside is a setting master and an alternative gauging unit. Constructionally, it resembles the fixture in the sectional drawing Fig. 19. From this it will be seen that the slender carbide-tipped contact plunger controls the lift of a ball that obturates the air nozzle. In the top of the gauging head, faced to 0.002 in below the gauging

Fig. 20. Multi-dial instrument for checking camshaft journals



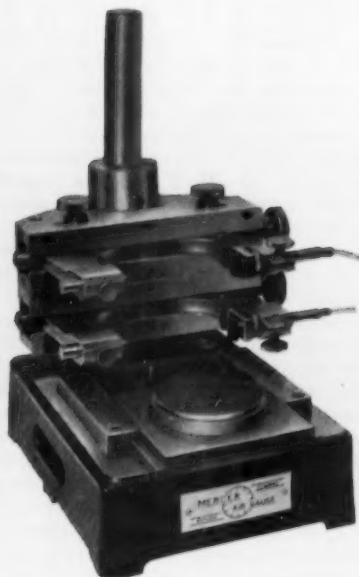


Fig. 21. Pistons of six different diameters are checked at two levels on this adjustable fixture

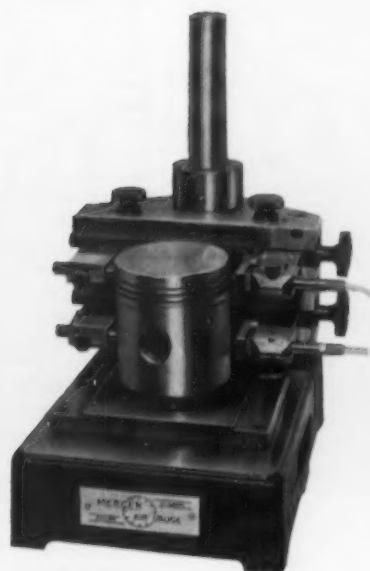


Fig. 22. The piston is slid into the gauging position between solid contacts and microjet units



Fig. 24. The M.I.1 precision comparator has a guaranteed repetitive accuracy of 0.00002 in

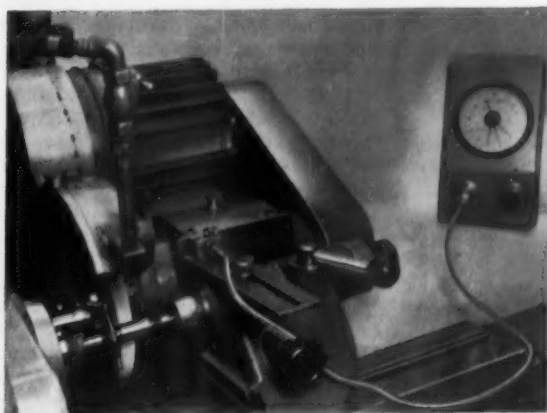


Fig. 23. Microjet used for fine control of in-feed on grinding machine

height, is a hard metal insert which prevents the plunger or the air nozzle sustaining damage in the event of an undersized component being forced down over the head. The height of the gauging head above the table is adjustable by a socket screw and sleeve nut device, laterally engaging a flange at the base of the gauging member.

Another application is the special fixture, Fig. 20, for gauging the three journals of the camshaft of a four-cylinder engine. This embodies three mechanical-contact gap gauges, similar to that shown in Fig. 16, used in conjunction with a typical multi-dial instrument.

Microjet plunger units

These small, cylindrical, mechanical-contact type gauging units are produced for incorporation in virtually any type of gauging fixture, for machine control applications, and as

measuring elements for comparators. The carbide-tipped plunger operates an obturating device which can be arranged to produce an amplification factor similar to, or greater or less than that obtained with a direct air jet. The amplification factor of a particular microjet unit is not adjustable.

Standard ranges are manufactured to different diameters and in long and short versions. To special requirements they can be made down to 0.3125 in diameter and 1.25 in long. There are two main types; normal acting and reverse acting. Depression of the plunger in the normal type, intended for external measurement, produces a pressure drop in the air system. The converse applies in the case of the reverse type, used for internal measurement.

Microjet units are shown mounted on the right-hand arms of the fixture for gauging land and skirt diameters of pistons, Figs. 21 and 22. The piston is located on a spigot plate slidable in guides on the fixture base and is then pushed forward to a stop to bring it into the gauging position. This

Fig. 25. Workshop air comparator fitted with microjet measuring unit

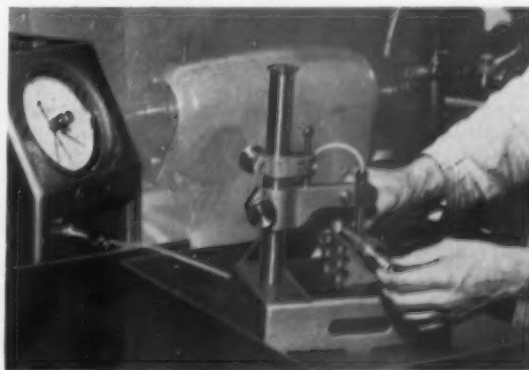




Fig. 26. Multiple dial-gauging set-up for flanged wheel hubs

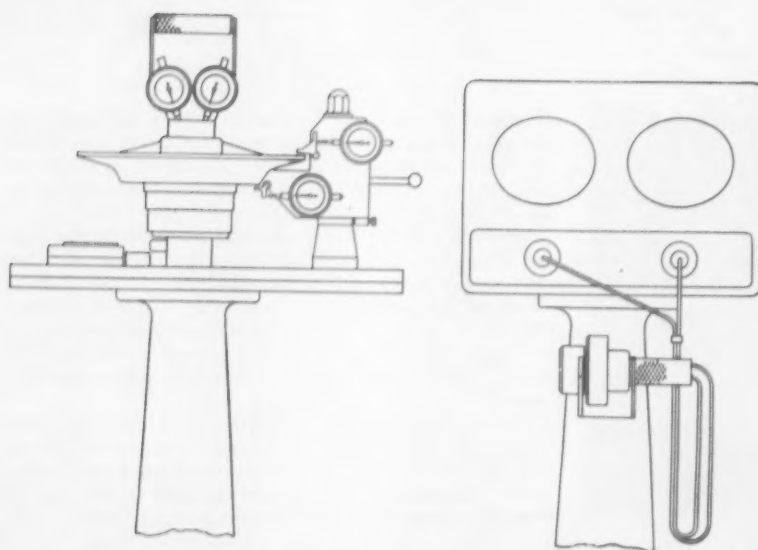


Fig. 27. Arrangement of gauging station for commercial vehicle wheel hubs

fixture is fully adjustable to accommodate pistons of six different diameters. Measurements are indicated on a twin dial instrument.

An instance of the unobtrusive manner in which a microjet unit can be fitted to a machine tool is shown in Fig. 23. This grinding machine is used for taking off fine diameters to finish work to very close tolerances. The microjet unit is mounted on a bracket secured adjustably to the tailstock on the work table and makes contact with a precision ground plate fitted to the front of the grinding head. A small standard instrument having a balanced scale range of 0.002 in, in increments of 0.0001 in is used. As the finishing cuts are taken a continuous indication of work diameter is given on the instrument and a close and sensitive control of in-feed of the wheel can be maintained. In the example shown, the work in hand was limited at ± 0.00002 in and no difficulty was experienced in holding work to the tolerance.

Air Comparators

These instruments, designed for workshop, inspection department and metrology department use, also embody a mechanical-contact microjet as the measuring unit. Two models are produced; W.S/1 expressly for gauging alongside the machine and for routine production checking and M.I/1 to meet the more exacting requirements of tool, inspection, and metrological departments. The M.I/1 model has a heavy base, a shrunk-in column of relatively large diameter, and a substantial arm to secure the necessary rigidity and stability to ensure consistent accuracy of measurement. Coarse and fine adjustment of the height of the measuring head is provided; the fine adjustment being on the column for W.S/1 and on the microjet mounting for M.I/1. These comparators are usually set up by means of slip gauges to the required tolerance. For normal shop use it is customary to supply setting masters for high and low limits. These may take the form of specially made slip gauges or of masters similar in shape to the component being measured. A repetitive accuracy of ± 0.00001 in is readily achieved. Platens of various types are produced to meet different requirements. That shown on the M.I/1 comparator in Fig. 24 is reversible; the two lapped surfaces being respectively flat and striated. The base can be furnished with bushed holes for the attachment of auxiliary equipment.

Combination gauging systems

While pneumatic gauging methods can be applied to most classes of work and are being more generally adopted, they do not in practice supersede other methods, either mechanical or electrical. Rather they offer the production engineer an additional and markedly different approach to gauging and inspection problems. Many factors such as the specific characteristics of the work, shape, size, accessibility, production rate, and the standard of precision specified, will influence the choice of method. The decision may also be affected according to whether gauging is to be performed on the machine, in the shop, or in an inspection department; whether gauging is for conformance to tolerance or for selective grading; and whether skilled or unskilled labour is to be employed.

The makers of the air gauging instruments illustrated in the foregoing pages, Thomas Mercer Ltd., Eywood Road, St. Albans, Hertfordshire, have for long manufactured a comprehensive range of mechanical dial-gauging equipment. In numerous instances mechanical and pneumatic gauges of Mercer manufacture supplement each other in the same plant, the same shop, or even on the same component. Figs. 26 and 27 relate to a recently supplied gauging station for commercial vehicle front wheel hubs. Two bores are gauged simultaneously by a two-diameter air plug gauge and a pedestal-mounted, twin-dial instrument. The scales are provided with coloured sectors which define the tolerances for easy and rapid scanning. After the air gauging, the component is placed on the revolvable locator of a closely adjacent, pedestal-mounted fixture. On this are checked the squareness of two abutment faces, the concentricity of two bearing diameters, and the squareness of two large diameter flanges. All these checks are made with mechanical dial gauges.

Investigation of Cutting Tool Life

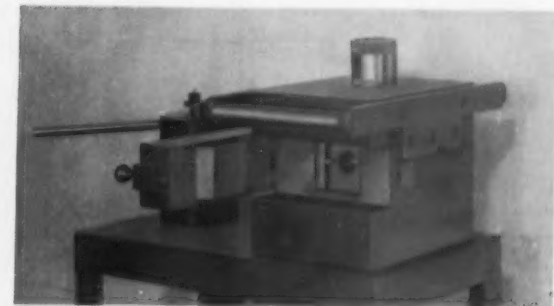
The Use of Radio-Active Tracer Elements to Determine Factors that Affect the Rate of Wear of Cutting Tools

BECAUSE of the current trend towards the employment of more and more complicated and expensive machine tools, idle time is becoming increasingly important as a factor affecting production costs. As a result, most manufacturers are constantly seeking means of increasing cutting tool life. The benefits of such increases are as follows. A larger ratio of production time to idle time of the machine tool is obtained. Replacement costs are less, so far as cutting tools are concerned, and less time is wasted in setting up and sharpening tools. The finish of the components is generally of a higher standard, and the life of the cutting fluid and its contamination rate is reduced.

Certain standard tests to evaluate cutting tool life, machineability of metals and effectiveness of coolants have been established for a long time. They involve drilling and turning under production conditions, and generally are lengthy and expensive since, in many instances, they have to be run till the cutting tool fails. Moreover, because of the long time and large amount of material needed for such tests, not only is the expense of carrying them out high, but also there is considerable likelihood of the introduction of variables that may lead to inconsistent or erroneous results.

The new tests

Alexander Duckham and Co. Ltd. claim to have the first laboratory in Great Britain for using radio-active tools for research into tool life, cutting fluids and machinability of metals. The basic principle of the tests is as follows. A radio-active cutting tool is employed; as it wears, radio-

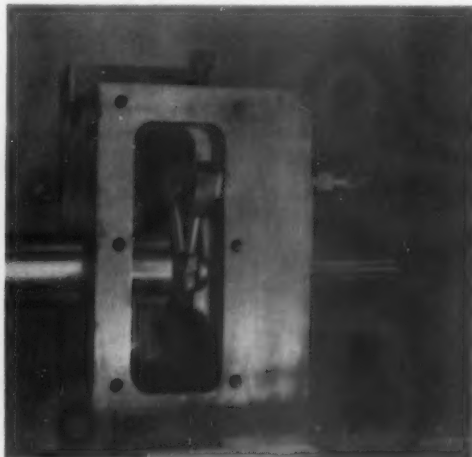
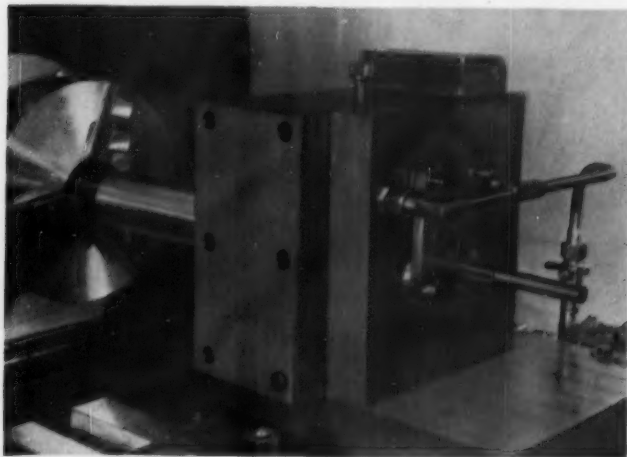


The tool tips are stored on a mandril inserted from above into a lead chamber. In this illustration, the door of the chamber, through which the tool holder is inserted to remove the tip, is shown open

active particles are transferred to the turnings and the cutting coolant. It has been established that 95 per cent of the tool wear products are always embedded in the chips, and the amount of radio-activity so transferred, therefore, bears a direct relationship to the wear on the tool. The amount of radio active material transferred can be determined by carefully drying and weighing the chips and exposing them to a scintillating counter, which records radio-activity as so many pulses per second on a scaling unit. This method of testing has the advantage that it is simple, rapid, and the amount of metal of the work piece consumed is only about a quarter of a pound. Moreover, the results are accurate and can be repeated.

The first stage in the development of this method was to select cutting tools that can be rendered radio-active by irradiation in a nuclear reactor to produce suitable radio isotopes. Earlier workers established that selected tungsten carbides treated in this way give tracer radio isotopes cobalt 60 and tungsten 185, which have adequate energy and a half-life long enough for the cutting tool to be satisfactory for studies over a reasonable period of time. The half-life is defined as the time during which an isotope loses 50 per cent of its radio-activity.

To protect the operators from radiation, the cutting tool and the end of the workpiece are enclosed in a thick steel chamber. On the left, the chamber is shown closed, ready for operation, and on the right it is open to show the two cutting fluid jets above the tool tip and holder





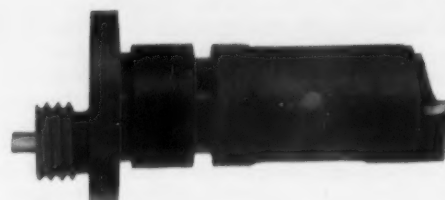
For the experimental work, Wimet X.8 tungsten carbide was selected. This material has the following chemical analysis: tungsten carbide 77 per cent, cobalt 8 per cent, titanium carbide 15 per cent. When subjected to neutron bombardment in a nuclear reactor for approximately one week, this material gives tracer radio-active isotopes of cobalt 60 and tungsten 185 with properties that were regarded as satisfactory for the initial studies. Cobalt 60 has a half-life of 5.2 years and emits beta and gamma radiation, while tungsten 185 has a half-life of 73 days and emits only beta radiations.

Tools

The tools used are approximately $\frac{1}{4}$ in. square by $\frac{1}{2}$ in. thick and weigh about 15 grammes. Each is finish ground to shape and marked for identification and has two cutting edges. The tools are sent to Harwell for irradiation and are hotted up until their total beta and gamma activity is approximately 15 millicuries per gramme of tool tip. One curie is defined as 3.7×10^{10} disintegrations per second. Of the 15 millicuries, 10 are of beta radiations and 5 are of gamma radiations. For the purpose of tool wear studies, gamma radiations only are used, because beta radiations have only slight penetration power and do not record well because of chip piling. Gamma rays, on the other hand, have good penetrating powers, since they consist of high energy rays

A long handle is attached to the tool holder to reduce the risk of exposure of the operators to radio-activity emitted from the tip. To actuate the push rod that releases the tool tip, the handle slides in the adaptor that is mounted on its end and screwed on to the tool holder

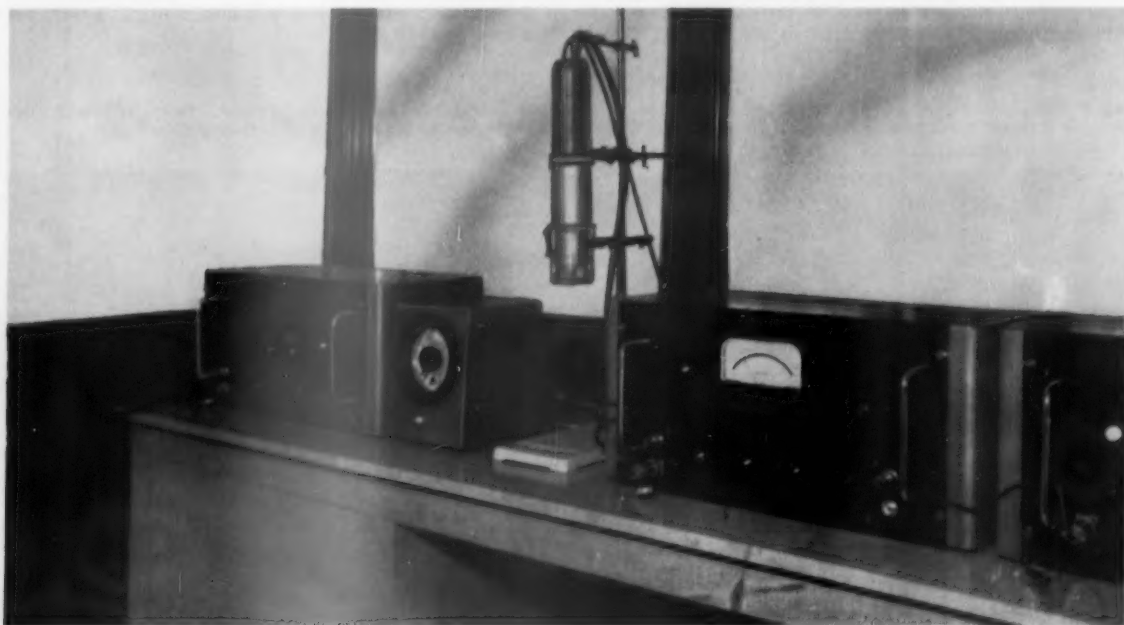
An enlarged view of the tool holder, showing a tip clamped between the jaws at its right-hand end

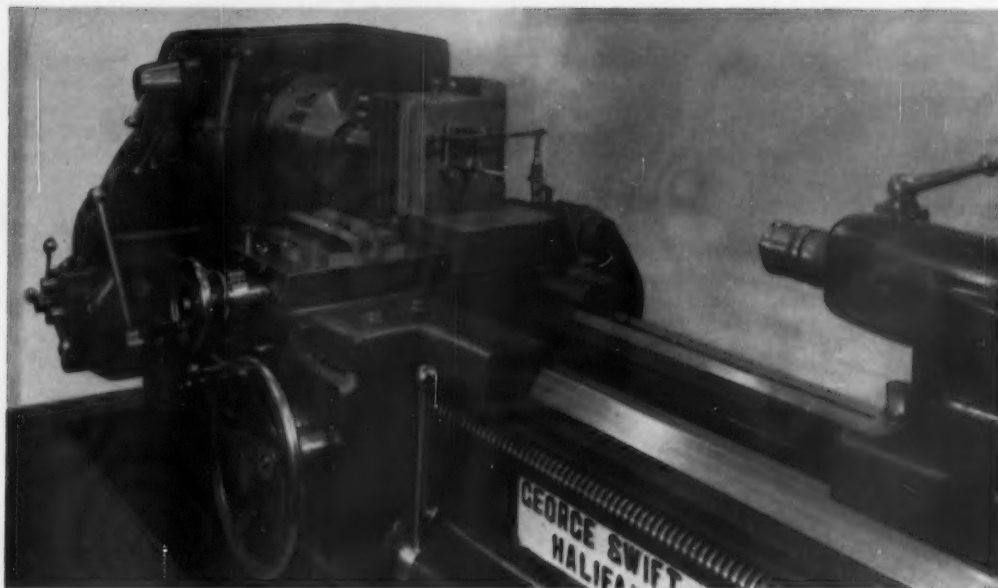


of short wavelength, and they also have good recording properties. Cutting tools activated in this way are estimated to give about 80 minutes of cutting time per tool edge.

It is only necessary to cut for a duration of 6-10 seconds to give gamma radiation counts of 3-5 per minute on the

The beaker of dried and weighed radio-active turnings is exposed to the head of the scintillating counter, and the radio-activity of the chips is recorded as so many pulses per second on the scaling unit. The beaker and head are near the centre of this illustration





The cutting chamber is mounted on the bed of a Swift BV.3 lathe powered by a 15 h.p. motor. Rigidity is the most important requirement for machine tools for tests of this kind, if inconsistent results are to be avoided

scintillator. Thus, once the machine has been set up, about 500 tests can be run off. It is normal practice to repeat each test five or six times to obtain an average of the results. In each instance, the results are checked against a standard. The tool wear is expressed as counts per minute per gramme of chips. It is necessary, of course, to deduct from this count the background radio-activity reading due to radiation naturally present in the atmosphere and also due to scatter from the radio-active material in the laboratory.

Safety precautions

In these early experiments, elaborate precautions have been taken in respect of the danger of nuclear radiation, although the activity of 15 millicuries per gramme is not high. The section of the mechanical test laboratory devoted to radio-active tests is separated from the remainder by metal and glass partitions, and is equipped with plastics covered benches and floors. These materials are impervious, so they can be cleaned readily and can be decontaminated quickly, should they come into contact with radio-active substances. Protective clothing and shoes are worn by all personnel working in this laboratory. A carefully designed and constructed decontamination zone has been established, through which all personnel must pass before leaving the danger area.

All personnel working in the laboratory are subjected to a medical check every five weeks, and stringent measures to ensure safety are adopted. Each operator is supplied with a personal dosimeter, which indicates the radiation risks that he is running and enables him to see at any time what he can or cannot do without exposing himself to harmful quantities of radiation. A log is kept of the degree of exposure of each operator, to ensure that he is not subjected to a dosage rate of more than 0.3 roentgens per week, which is the maximum permissible.

The radio-active cutting tools are stored in a specially constructed lead chamber, in the centre of which is a mandril to hold four tool tips. This mandril can be rotated so that the tool holder can select any one of the four tips. The tool holder is inserted from a safe distance and a spring release cam is operated to engage the tool tip. Then the tool is withdrawn from the chamber, and the lid closed.

Apparatus and method

In the initial experiments, tubular steel work pieces were employed. The material was a 60 ton chromium molybdenum steel in the annealed condition. Cuts were taken normal to the major axis of the tube so that the operation could be carried out in a steel chamber, which surrounds the tubular work piece, held in the chuck of the lathe, and screens the operators from radiation. Tubing was also chosen for the work piece because it avoids problems with regard to surface condition, and also a constant cutting speed in terms of ft/min can be obtained readily.

The cutting chamber, the walls of which are 1 in thick, is mounted on the bed of the lathe. It is designed so that access for changing the tools is easy and also so that it can be cleaned readily. A cutting fluid circulation system is incorporated in the chamber. It supplies cutting fluid through two jets, one on each side of the tool. The fluid drains into the base of the chamber, is collected into a reservoir and then recirculated over the work piece and tool. All the turnings are collected in a small perforated tray.

Cutting proceeds until approximately 1½ oz, 50 grammes, of chips are produced. This has been found enough to give an adequate radio-active count. In the tests carried out so far, the time required for machining this amount of metal is 10 sec. The machine setting for speed and feed can be varied according to the type of study. Speeds of 200-800 ft/min are possible with feeds from 0.003-0.15 in per cut. When the cutting operation is completed, the perforated tray containing the chips is taken from the chamber. The chips are then placed in an aluminium or stainless steel tray, taken to the measuring laboratory and washed. Water and methylated spirits are the washing agents for a soluble oil coolant. When a neat oil coolant has been employed, solvent naphtha is used for washing. Finally, the chips are dried, weighed and exposed to the scintillating counter for measurement.

Alexander Duckham and Company express their thanks and appreciation to the Directors of Research, Ernst and Merchant, of the Cincinnati Milling Machine Company, Cincinnati, Ohio, for their advice and guidance in the planning of this new research laboratory.



Automobile wheel with eight spot welds has a shear strength of 45,000 lb

SPOT-WELDED WHEELS

Sciaky Automatic Three-phase, Resistance Welding Machines



This heavy-duty truck wheel has twelve spot welds of 0.75 in diameter

EARLY attempts to produce spot-welded automobile wheels met with indifferent success. Difficulty was encountered in maintaining consistency in the welds and, understandably, wheel manufacturers were reluctant to change their method of production from riveting to welding. It would appear that the lack of consistency arose from wide variation in the welding heat. Even when a good setting had been established, excessive heating and expulsion of metal was likely to occur after some time in production, and with continued production welds would become "cold", with lack of penetration and reduced nugget size. This was found to result from the changing contour of the electrode tips in various stages of deterioration. The effective working life of the electrode tips was short. In endeavours to obtain greater consistency, machines were built in which the electrodes were regularly dressed and re-contoured, and automatic machines were equipped with mechanisms to alternate the electrodes. Such devices were merely palliatives, however, and left the basic problem unsolved.

Three-phase welding machines

It was the advent of three-phase machines that made the spot-welding of wheels practical. Further investigation of the problem revealed that electrode life was shortened by the skin effect due to the sharp rate of rise of the conventional supply frequency of 50 or 60 cycles. This characteristic caused the current to be concentrated around the periphery of the electrode rather than to be evenly distributed throughout the cross-section. The consequence of this was "mushrooming" of the tip and variation of the tip contour and contact area. Pulsation welding gave only a slight increase in electrode life with the conventional supply frequency. The improvement was probably due to enhanced cooling of the electrode during the "off" times, but the characteristic skin effect, and its consequences, still obtained.

Electrode life was enormously increased by the use of fully rectified three-phase welding current. The secondary current consists of low-frequency alternating pulses having a slow rate of rise. Skin effect was virtually eliminated and heat distribution over the electrode area became uniform. The off time between each current impulse allowed time for the heat to travel throughout the weld zone of the work and

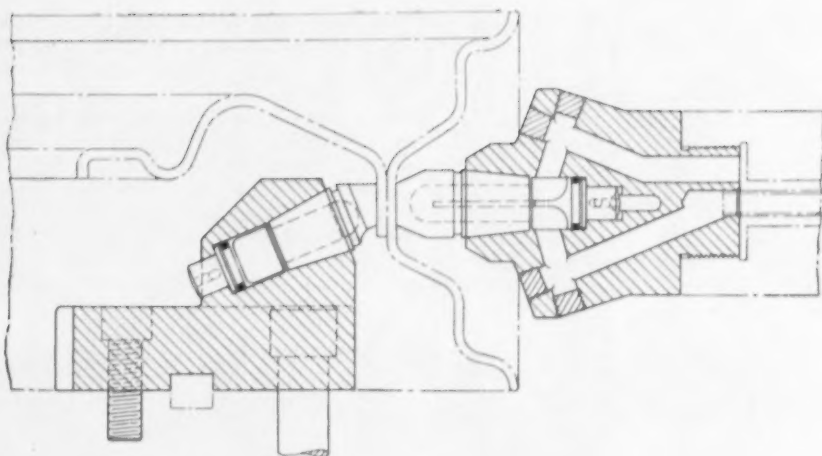
cooling time for the electrode. It became relatively easy to maintain the contour of the electrode.

The three-phase system conferred other advantages. Its characteristic of adjusting the secondary current automatically for variations in secondary resistance reduced the shunting of current through previously made spot welds. (Additional current is demanded to replace that lost in shunting.) This characteristic also compensates for variation in work thickness, chemical composition of the work material, and for some variation in tip contour and contact area.

Special attention was given to the design of the internally cooled electrode tips to improve the heat transfer. A round-nosed drill was used in boring the water chamber and a flat baffle plate was fitted therein. This ensured that the incoming water travelled right out to the end of the tip and prevented air bubbles being trapped in the electrode. The chamber was bored to a depth that left a wall only $\frac{1}{16}$ in thick between the weld and the cooling water. Improved cooling of the tip substantially increased the working life of the electrode and an average life is now about 60,000 wheels, that is 120,000 welds.

Wheel design considerations

It is of importance that the mating parts of the wheel are



Electrode assembly showing special cooling arrangement to extend tip life

a good close fit in order that the contact area is consistent and the full electrode force is applied to the weld. Commonly an interference fit of the order of 0.040in is arranged between the spider and the rim, and has proved to be satisfactory. The width of the spider flange is increased somewhat beyond that usual in riveted assemblies to provide adequate material for the relatively larger welds. It is desirable that the flat width of the flange should be about 1.0in, with $\frac{1}{8}$ in as a minimum. This can be arranged without increasing the area of the blank if less material is trimmed from the corners of the blank. The corners of the spider arms are washed out by appropriate shaping of the pressing die so that the press fitting of the spider in the rim will not merely result in local contacts at the corners.

In riveted assemblies it is usual to use twelve rivets.

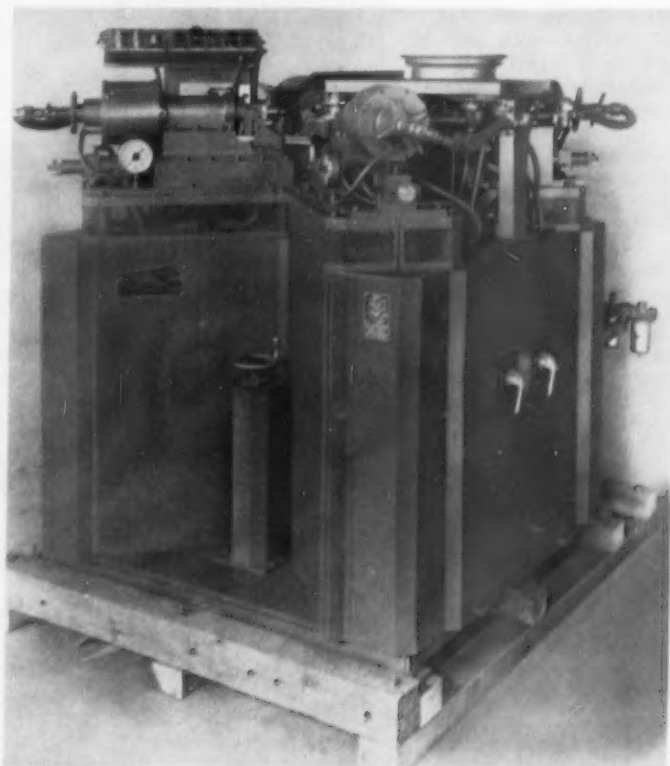
Extensive investigation, experiment and testing established that three-phase, low-frequency welding could produce consistently exceptionally large spots of excellent metallurgical quality and high shear strength. Thus it became possible to limit the number of welds to eight spots per wheel, giving increased economy. In a wheel having a rim thickness of approximately 0.125in and a spider approximately 0.110in, eight spot welds having a minimum nugget diameter of 0.500in had a shear strength of 45,000 lb whereas the same wheel assembled with twelve rivets sheared at 38,000 lb.

Early wheel welders were merely standard general-purpose machines equipped with appropriate electrodes and holders to suit the wheel configuration. These machines made only one spot at a time and the wheel was supported on a fixture and indexed manually to obtain the required number of welds. While the welds were of the stipulated high quality, the rate of production was necessarily limited. Not more than 110-120 wheels per hour could be welded on these machines. Accordingly, special automatic machines were designed for installation in the produc-

tion line and by making multiple welds the rate of production was considerably increased.

Automatic wheel welders

The first machine for the mass production of spot-welded wheels was the Sciaky MGT-4. This is a table type machine with four sliding, air-operated, direct-acting welding guns mounted horizontally on the top of the machine and arranged at 90 deg to each other. It forms a complete production unit, including automatic indexing, automatic unloading, and welding control, and is capable of handling wheels of from 14in to 20in diameter with eight spot welds. No manual operations are required after the pre-assembled rim and spider has been loaded and located until the completely welded assembly is automatically unloaded. A washing



Sciaky MGT-4, automatic indexing, four-gun, wheel welding machine

operation precedes the assembly of the rim and spider. The bath includes a rust inhibitor so that the wheels may, if necessary, be stored before welding.

Each of the four welding guns has its own three-phase transformer housed in the box frame of the machine and enclosed by sheet metal covers. All transformers are sealed as a protection against possible water leaks from the cooling circuits above. The gun is attached to a water-cooled, copper alloy yoke mounted in gib slides and is operated by a horizontal air cylinder. A second piston in this cylinder can be positioned by a hand wheel to set and adjust the stroke of the ram to suit the depth of the rim and to minimize air consumption. A small air cylinder is also provided to retract the inner electrode while loading and unloading are carried out. The inner electrode is mounted in an adaptor dowelled and bolted to the gun yoke, and ring type seals are fitted at the drillings for the coolant, so no external connections are required. As a maximum the gun has a welding stroke of 3½ in, and an electrode force of 5,000 lb can be exerted with an air line pressure of 80 lb/in². The initial position of the yoke is readily adjustable to accommodate wheels of different diameters.

An elevating platform of a non-magnetic material lowers the wheel from the loading station over a central locating pilot and the wheel is then automatically clamped against a shoulder on the pilot. One of the wheel bolt holes

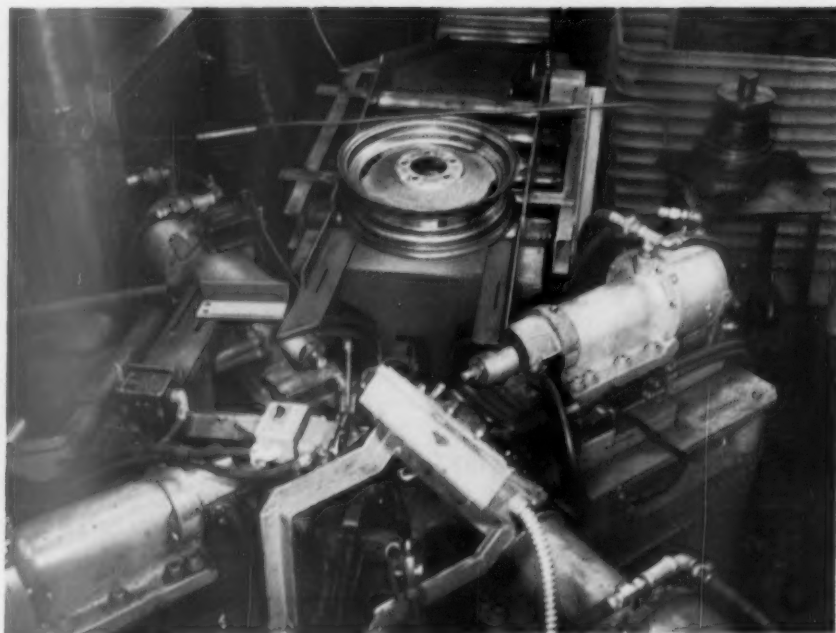
is engaged over a pin on the pilot to ensure correct radial location with respect to the welding electrodes. This pin also serves in connection with the later indexing operation. The platform is mounted on a carriage sliding on vertical guide posts and is operated by an air cylinder, the stroke of which is adjustable to suit rims of different widths.

After a group of four welds has been made the wheel is automatically indexed by means of a double-acting air cylinder. The stroke of this component is adjustable so that spider arms of various widths can be accommodated. The automatic unloading device consists of a double-acting air cylinder mounted on a superstructure above suitable



Arrangement of four sliding welding guns on the table of the MGT-4 machine

Overhead view of MGT-4 machine with elevating loading conveyor and lateral unloading equipment





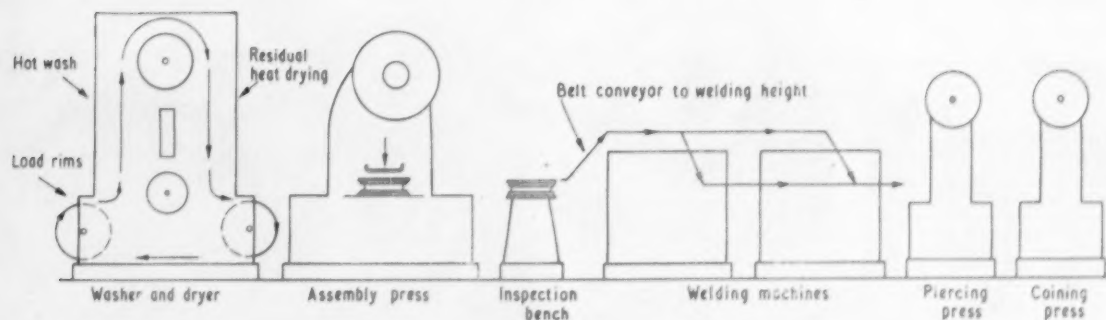
Sciaky MGT-8 fully automatic transfer machine for the production of 700 wheels per hour

guide rails. When the completely welded wheel is raised to the unloading position the unloader pulls it clear of the welding machine and then returns after the succeeding wheel has been lowered to the welding position.

Mechanical functions of the machine are initiated by push buttons and limit switches, with suitable cut-out controls for emergency stops. In automatic operation the sequence of the machine is initiated by loading a wheel against a limit switch. The initial radial location of the wheel when loading is the responsibility of the operator. The wheel is fed to the machine on a smooth surface, guided by rails, on to the platen where it is pushed to a stop, which also forms a rail for guiding the wheel off the machine. In position against the stop it actuates a limit switch which causes the platen to lower the wheel to the welding position. Three limit

switches arranged in series check that the wheel is in the correct welding position and is horizontal. The closing of all three switches actuates the holding-down clamp and causes the welding guns to advance. Four welds are then made simultaneously in the corners of the spider arms and the guns are retracted. The indexing mechanism turns the wheel into position for the next group of welds in the opposite corners of the spider arms and the guns are again advanced. At the end of the second welding operation the guns are retracted and a limit switch causes the platen to raise the wheel to the unloading position and immediately the unloading device pulls the wheel clear of the machine. The feeding of the succeeding wheel into the loading position restarts the sequence and when this wheel is lowered over the centre pilot the unloader commences its return stroke.

Schematic layout of production line for 770 wheels per 60-minute hour



The welding transformers, nominally rated 125 KVA at 50 per cent duty cycle, are specially designed and constructed to operate efficiently and without overheating at the exceptionally high duty cycle at which the machine is operated. It demands 160 KVA per gun, or 640 KVA for four guns operated at the same time. A Sciaky three-phase control of the synchronous electric type is provided to fire the four transformers simultaneously. The control includes squeeze, weld interval, synchronous heat, synchronous cool, and hold timers, and also a phase shift heat control. Integrating the various functions of the machine, as well as the control of the welding cycle, is a master sequence control. Interlocks are provided to avoid the possibility of accident. A typical weld setting is 4 cycles of heat, 4 cycles of cool, 9 impulses at 90 per cent phase shift, and an electrode force of 4,000 lb. One customary practice is to use 4 cycles of heat and 4 cycles of cool for all sizes of wheel, varying only the number of impulses, phase shift, and pressure for different wheel thicknesses.

The optimum operating cycle of the MGT-4 machine, as analysed by time study, is as follows:

Function	Cycles
1. Limit switch makes; air exhausts from lift cylinder	20
2. Platen lowers to welding position ..	17
3. Welding guns move in; contact gauge makes	22
4. Squeeze time	16
5. Weld time, 13 impulses, 4 heat—4 cool	106
6. Hold time	33
7. Welding guns retract; wheel is indexed; guns move in	48
8. Squeeze time	16
9. Weld time, 13 impulses, 4 heat—4 cool	106
10. Hold time	33
11. Welding guns retract; platen rises to unloading position	48
12. Wheel unloaded	30
Total time, machine	495=8.25 sec
Loading time, assumed	0.93 sec
Total time per wheel	9.18 sec
Total production is thus 392 wheels per 60-minutes hour or 326 wheels per 50-minute hour.	

Typical production line

A typical production line, set up for a rate of 770 wheels per 60-minute hour is illustrated schematically. It comprises a conveyerised washing and drying machine for the rims, an assembly press, an inspection bench, two MGT-4 welding machines, a press for piercing the valve hole and a press for coining the rim to receive the decorative wheel covers. The line is separated from the rim rolling line and the spider



Testing the strength of a welded wheel by forcing the spider out of the rim and tearing out the weld slugs

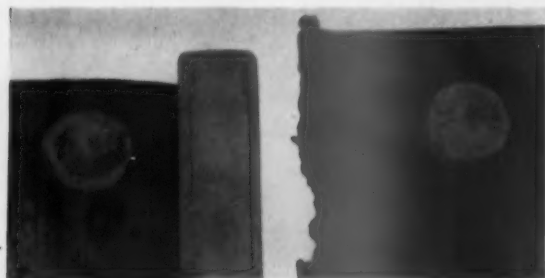
production line. Eight men are required to run the line. One man loads the rims on to the washer conveyor, the rims being hot-washed while travelling vertically up one side of the machine and dried by residual heat while being returned down the other side. Three men service the assembly press. The first takes the rim from the washer and loads it into the press; the second takes the spider from a rack and places in the rim on the press; the third takes the wheel assembly from the press and checks it on the inspection bench. The two welding machines and the two presses are each operated by one man.

All electrodes are checked every 55 min. and re-faced, if necessary, every 4 hr. Average life of the electrodes is 60,000 wheels for outer tips and 40,000 wheels for the inner tips. The nugget diameter is maintained at not less than 0.500 in.

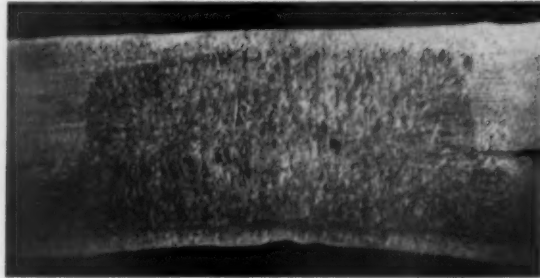
Automatic transfer wheel-welding machine

For a higher rate of production the fully automatic MGT-8 machine has been developed. In essentials, this comprises two inverted MGT-4 welding machines connected by a transfer conveyor system. Each of the welding units makes four of the eight welds on each wheel. The conveyor carries the wheel through the entire sequence of operations, including pressing the spider into the rim, welding the wheel

Typical welds cut from a wheel and then sheared under test. Minimum nugget diameter is 0.500 in



Weld cut from a wheel and cross sectioned and etched for metallurgical inspection



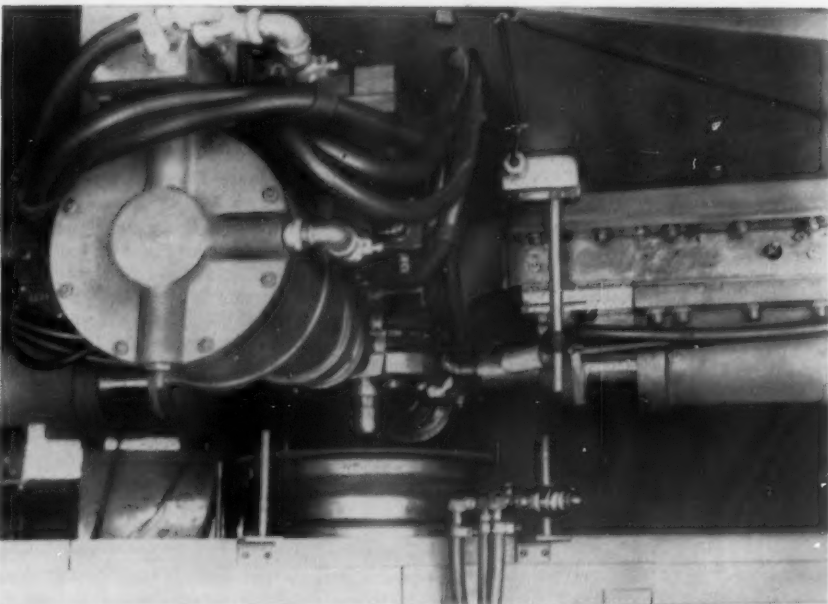
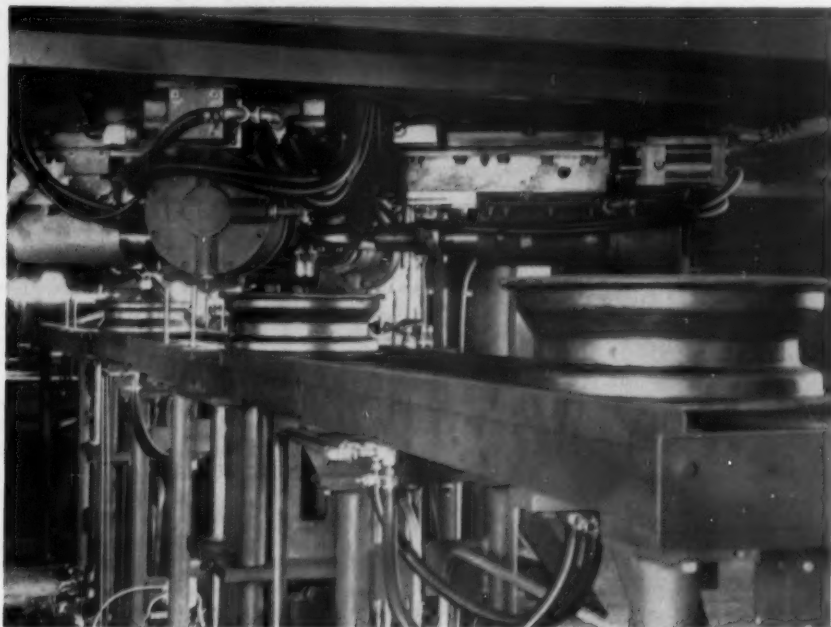
assembly, automatically checking the quality and consistency of all welds, and punching out and dimpling the valve hole. The two welding units are each of four-post construction with fabricated steel base plates and crown plates suitably reinforced to minimize deflection. On the crown plate are suspended the sliding welding guns and above are supported the welding transformers. The guns are mounted on an adaptor plate that provides adjustment of the angular relationship of the two groups of guns between 30 deg and 45 deg, to vary the spot location for different sizes of wheels and for different spiders. Construction and mounting of the welding guns follows the arrangement on the MGT-4 machine. On that machine the wheels were carried over

the guns and lowered into the welding position but on the MGT-8 the wheels are transferred below the guns and are raised to the welding position by an air cylinder and guiding mechanism. After welding, the wheel is lowered back on to the conveyor for transfer to the next station.

At the pressing station the spider is fitted into the rim by means of a 12in diameter air cylinder. The pressing dies are designed for easy and rapid removal and replacement when changing over the tooling for wheels of different sizes. The alignment and location of the component parts are effected by the centre pilot and the dies themselves. This is of importance, since once the wheel is assembled and welded, run-out and misalignment cannot be rectified. Radial

orientation of the wheel, to ensure the correct location of the arms of the spider with respect to the welding guns, is accomplished on the transfer mechanism.

The last two stations of the machine are for punching and dimpling the valve hole in the rim. Each of these press units consists of a 12½-ton hydraulic cylinder fitted to a sliding yoke and loaded by heavy springs so that the stationary die clears the rim during transfer operations. They are mounted to apply the press force normal to the appropriate surface of the rim and are fully adjustable to handle wheels of different diameters and rim sections. The guide rails for the wheel at these stations engage the tyre-bead seat of the rim. Thus, after punching and after dimpling, the wheel



On the MGT-8 machine the conveyor carries the wheels below the suspended welding guns

Every weld is examined by a photo-electric cell and a defective weld is automatically indicated by a shot of paint

is held down while the tools are stripped from the work under the action of the loading springs. The hydraulic power unit furnishes a 1,000 lb approach pressure and a 5,000 lb operating pressure.

Much thought and development has been given to the transfer mechanism which must hold the wheels firmly and move them rapidly and precisely in order to maintain the rate of production. Approximately 30 cycles, or 0.5 sec, are required for a wheel to be grasped and transferred from one station to the next, a distance of about 19in.

Automatic weld checking

Each individual spot weld on the wheel is automatically inspected and evaluated by the Sciaky weld checker. This device examines the weld immediately after it has been made and if its quality does not conform to a predetermined value it is automatically indicated with a shot of red paint so that later the inspector can locate and examine the wheel. The weld checker utilizes a photo-electric cell unit in conjunction with a bridge circuit to test the quality of the weld. Infra-red radiation from the completed weld energizes the photo-electric cell and thereby brings the bridge circuit into balance. Should the weld be too cold or too hot the radiation will be too little or too much and the bridge circuit is unbalanced. In such circumstances a relay is energized to start a sequence that delivers a shot of red paint from an appropriately mounted spray gun. The checker may also be equipped with visible or audible signal devices to inform the operator when a questionable weld has been made, and thus minimize the production of sub-standard components.

Wheels having a weld that does not conform to standard are set aside, examined, and in suitable cases can be recovered by adding a weld close to the sub-standard spot. This work is done on a standard single-point welding machine. The checker is adjustable to the desired heat required for a high quality weld on a specific wheel and is quite sensitive to variations from the pre-set value.

The operational sequence of the machine is:

1. Rim automatically loaded on conveyor
2. Spider manually loaded on rim and oriented with respect to the welding guns
3. Spider pressed into position
4. Four welds made in first welding machine; welds checked
5. Wheel transferred to second welding machine; four welds made; welds checked
6. Valve hole punched; hole dimpled
7. Unloaded

The sequencing is arranged so that the first and second welding machines are operated at different times in order to reduce the power demand, but in such manner as to avoid any loss of time in the operating cycle. When the second machine is welding, the first machine is moving a wheel into position by raising it up to meet the electrodes and moving the welding guns into contact with the wheel preparatory to welding. While the first machine is welding, the second machine retracts the welding guns from a completed wheel, lowers the wheel to the conveyor and, as the wheel is moved out to the valve hole punching station, the next wheel to be welded is moved into position ready to be raised to the electrodes.

Nominally the MGT-8 machine is designed for the production of wheels at the rate of 700 per hour. That this rate can be comfortably exceeded is shown by the following study of the operating cycle:

Function	Cycles	Sec
1. Transfer movement	30	0.50
2. Platen rises	47	0.78
3. Guns move in; squeeze time	25	0.42
4. Weld time, 13 impulses, 5 heat—2 cool	91	1.52

5. Hold time	16	0.27
6. Guns retract	15	0.25
7. Platen lowers	52	0.87

Total time 276 4.61

At 100 per cent efficiency, that is, a 60-min hour, this figure gives a production rate of 780 wheels per hour.

Testing welds and wheels

Various destructive tests are employed by the manufacturers of the machines, Sciaky Welding Machines Ltd., Farnham Road, Slough, to evaluate the quality of welds and to compare the relative strengths of welded and riveted wheels. One of the tests is to use either a press or a tensile testing machine to extract the spider from the rim and then to inspect the welds. The manner in which the test is conducted results in a combined tensile and shear load on the welds. They are considered to be of good quality if all of them pull slugs of material out of the metal and reveal a weld nugget of at least 0.500in diameter, as shown earlier. Welds are also frequently cut out of a wheel and then sheared individually in a tensile testing machine or cross-sectioned, polished, and deep etched for metallurgical examination. Hardness checks are also made at various points in the weld nugget and the adjacent base material.

Destructive tests on complete wheel assemblies are made by a variety of methods. In one of these the wheel, complete with tyre, was supported at an angle of 30 deg from the horizontal and a weight of 2,020 lb was dropped upon it from heights of 1 ft, 2½ ft and 3 ft. Rim, spider and hub were damaged beyond repair but the welds were not impaired. A similar wheel having a riveted assembly could withstand a drop of only 7in by the same weight.

In another test a wheel complete with tyre was clamped horizontally to the table of a Bullard machine and rotated at a speed equivalent to a vehicle speed of 20 m.p.h. A shaft 4 ft long was bolted to the spider of the wheel and at the end of this was imposed a lateral load of one-fourth of the vehicle weight with a 60 per cent overload, that is 1,040 lb. On this test a riveted wheel was considered satisfactory if it would survive 27,000 revolutions. The welded wheel withstood 189,000 revolutions before failure occurred at the spider bolt holes. The welds, however, were still in good, sound condition.

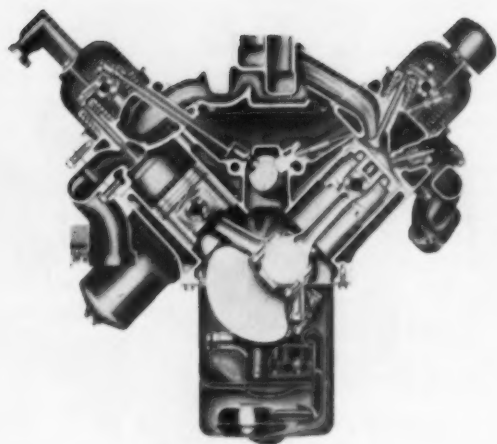
Mechanical Handling

Two special issues of our associated journal *Mechanical Handling* will deal with the 5th Mechanical Handling Exhibition and Convention, which is to be held at Earls Court, London, from 9th to 19th May. The first of the special numbers will be the May Exhibition Preview, and the second will be the June Exhibition Report. Both will be published at the normal price, 3/-.

The Preview number will be published just previous to the opening day of the Exhibition, May 9th. It will contain a full list of exhibitors, a plan of the Exhibition, and details of times and subjects to be discussed at the Convention. In addition it will contain all the normal features dealing with every aspect of industrial mechanization.

The Exhibition Report will be published on June 11th. It will contain details of the opening ceremony; a fully illustrated report of the equipment displayed; pictorial interviews with distinguished visitors; and a full Convention report.

In addition to their great topical interest, these two special issues will be of value as reference works. Copies can be obtained from all newsagents, or direct from Iliffe and Sons, Ltd., Dorset House, Stamford St., London, S.E.1.



Chrysler Plymouth V-8

V-8 ENGINES

A Review of Current Engine Designs in the United States of America

AT least nine out of every ten motor cars manufactured in the United States in the last year were fitted with V-8 engines. In fact, it is now nearly ten years since new tooling has been set up to manufacture any other type of automobile engine in that country; what few in-line engines are still required are produced on old tools. It is therefore worth while to review the advantages of the V-8 engine which have brought it almost universal use, to describe the engines now in production and to examine a typical engine critically from the points of view of design and all-round performance.

History

The first V-8 engine of which we have any record seems to have been the Antoinette aircraft and marine engine of about 1903. This was followed by a Rolls-Royce V-8 engine in 1905 and, a few years later, by a DeDion-Bouton engine of the same type. The chapter of engine development that interests us here may be said to have commenced with the Cadillac V-8 engine of 1914. This was the first engine of the type to go into series production and the only engine which—apart from a brief excursion into a V-12 and a V-16—has been under constant development ever since.

As originally produced, this engine had a single-plane crankshaft and therefore had a resultant secondary unbalance, a force which tended to move the engine horizontally from side to side. In spite of this, the 1914 Cadillac engine appeared to be extremely smooth and quiet in comparison with other engines of that day. In 1926 this unbalance was eliminated by the introduction by Cadillac of the 90-180-90-deg crankshaft, with the crankpins in two planes. With the change from side valves to overhead valves in 1949, this engine may be said to have started the move toward the universal use of V-8, overhead-valve engines in the U.S.A.

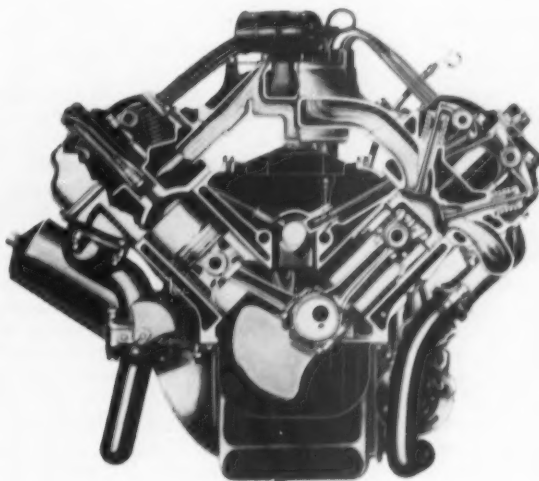
Another engine that deserves mention is the V-8 Ford engine, first produced in 1932. At the time, one wondered how an 8-cylinder engine could possibly be produced at a cost to compete in the low-priced field, yet this engine has been in constant production ever since and no less than 21,400,000 engines of the type have now been manufactured

by the Ford Company. Until 1949, however, Cadillac and Ford were the only large-scale manufacturers of V-8 engines in America. So the change-over to the V-8 type has been rapid and today, apart from lorry and tractor engines, there are indeed few petrol engines of any other type still in production.

Advantages

This universal trend toward the overhead-valve V-8 engine in America can be ascribed to two factors—the constantly-increasing anti-knock quality of the fuels available, now up to 100 octane, and the ever-increasing use of automatic machinery. This transfer-type machinery has made it possible to produce engine parts with finished surfaces in almost any location without cost penalty, provided that the cutting tools can be worked in without any great increase in the number of stations.

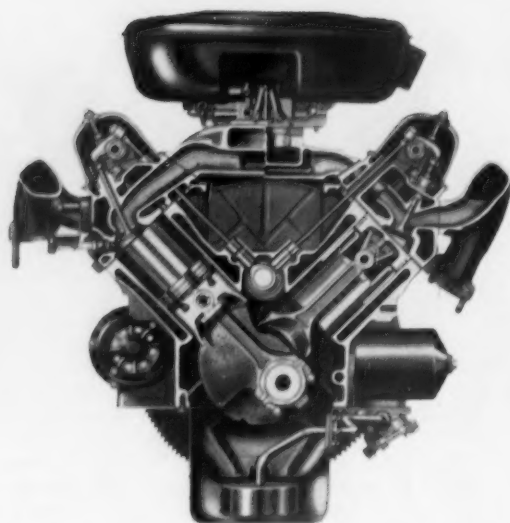
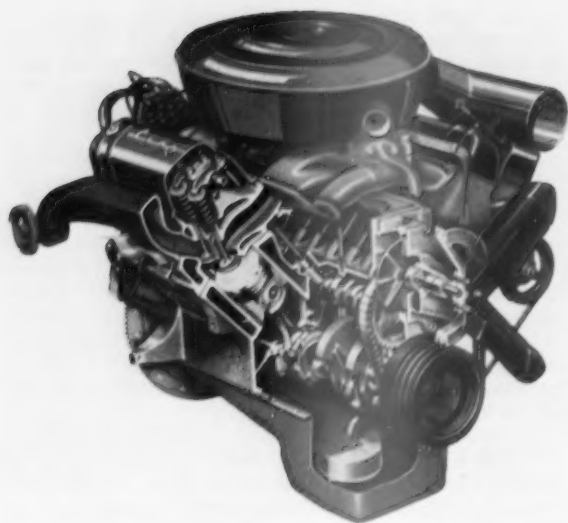
Investigation has shown that, of all possible engine types, the V-8 layout possesses the majority of advantages for smooth running at high compression ratios. Inherently it



Chrysler Firebird V-8

has a rigid crankshaft and crankcase. For a typical crankshaft, the 4th mode of torsional vibration will synchronize at about 5,000 r.p.m., while the deflection of the flywheel on account of the working load on the rearmost crankpin is exceedingly small.

It has been found that, for satisfactory operation at high compression ratios, an engine must have cylinder bores that stay round, rigid pistons and connecting-rods, durable bearings and high-voltage ignition. A considerable number of standard engines have been converted to compression ratios as high as 12:1, and have run very satisfactorily at these ratios. Incidentally, the V-6 engine has also been thoroughly investigated and a number of samples built, but this type has not proved so satisfactory as the V-8 engine.



Ford Lincoln V-8. Displacement 368in³

Power output

In considering the performance of present-day American engines, it will be convenient to set up a typical, or average engine. The reader may wish to compare this performance with that of the Chrysler V-8 engine of four years ago, which was described in the March 1952 issue of *The Automobile Engineer*.

From the following table, which includes all American V-8 motor car engines now in production, we may select a typical engine, somewhat larger than average, with a displacement of 350in³ (5.8 litres) and a bore and stroke of 4in × 3½in. With a compression ratio of say 9:1, this engine, bare, could be expected to give about 220 h.p. with a single, duplex carburettor and perhaps 280 h.p. with high-lift cams and two quadruplex carburettors. These figures correspond to 0.63 h.p./in³ and 0.80 h.p./in³ respectively and, assuming that they would be obtained at 4,600 r.p.m., would also correspond to 72lb/in² and 92lb/in² b.m.e.p. at that speed. The average of all the rated maximum torque outputs of these engines is close to 11lb-ft/in³, which corresponds, of course, to 151lb/in² b.m.e.p.

Now it will no doubt be pointed out that the horsepower figures given in the table are "advertised" figures, drawn on graph paper borrowed from the Engineering Department by the Sales Department, so that the engines may compete in the fantastic American "horsepower race". This is undoubtedly true, but the figures we have selected above are rather lower than the advertised figures and could, we believe, be obtained from a carefully-tuned engine under ideal conditions of intake air temperature, exhaust back-pressure and so on.

It must be remembered that, in each Engineering Department concerned, there has been, for some years past, a small group of engineers assigned continuously to the problem of obtaining more output from an engine to be manufactured on a given set of tools.

This unending search for an increased power output has made progress by taking much the same steps as the amateur racing, or "hot-rod" enthusiasts, including:—

- Increasing the cylinder bore.
- Fitting one or two four-barrel carburettors.
- Increasing the compression ratio.
- Increasing the intake port diameter.

Increasing the exhaust port diameter, smoothing the ports and easing the bends in the exhaust pipe; fitting two exhaust pipes, one from each bank of cylinders.

Increasing the valve-opening periods, increasing the cam lift or the rocker-arm ratio.

Fitting mechanically-adjusted tappets, where hydraulic tappets have given trouble at high speeds from "pump-up".

Lowering the engine friction.

Many of these changes can be considered as healthy developments, in that they quickly show up poor combustion-chamber design, springy valve gear, too light pistons and so on. That is, they tend to improve the breed. High power outputs have come in for much criticism on the part of those interested in safety on the road; they can be justified to an extent by pointing out that high torque at high engine speeds makes overtaking easier and thus possibly safer; furthermore, the engine has an ever-increasing accessory load, which now includes in most cases a hydraulic pump for the servo-assisted, or "power" steering and, in some cases, a compressor for the air-conditioning system.

Whatever the merits of the case, it is interesting that, at the beginning of this year, several manufacturers have offered, either as standard or as an extra, their highest-output engine for every chassis model in which it can be fitted.

A typical engine in this American V-8 group has a bore of 4in × 3½in and a displacement approximating to 350in³. The crankshaft is of forged or cast steel, with integral counterweights. The crankpins are 2½in in diameter and the main journals 2½in in diameter, so that the crankpins overlap the main journals by ½in. The main journals are of the same length, except the rearmost, which is usually twice as long as the others. Both the foremost and the rearmost crankshaft cheeks are almost twice as wide as the remaining cheeks, in order to minimize deflection at the flywheel and to ease the counterweight problem. A small torsional damper, either of the moulded-rubber or bonded-rubber type is usually fitted at the front end of the shaft. The flywheel is exceptionally light, being little more than a thin steel disc, plus attached parts of the automatic gearbox.

The length of the connecting-rod, centre to centre, is 6½in in an engine of this size, or 1.9 times the stroke. The connecting-rod cap is secured by two bolts, fitted with self-

locking nuts. The pistons are of aluminium, have flat heads and comparatively short skirts, which are cut away at the sides to clear the counterweights. They tend to become heavier in section as compression ratios are increased, and are fitted with two compression rings and one oil-control ring, all above the piston pin. The piston pin diameter is just under 1 in.

The design of the cylinder block follows a conventional pattern, the only controversial point being the location of the lower, or sump face, which may be either at or close to the plane of the crankshaft axis or several inches lower. The lower location is a feature of the Ford group of engines, which are described in advertising as "Y-8", rather than V-8 engines. The camshaft is located close to the centre of the block and is driven by inverted tooth chain in most designs, but in one engine by "Celeron" and steel gears. The distributor and oil pump are driven from the camshaft, towards its rear end, by spiral gears.

The cylinder head design is also fairly uniform, with variations in the angle of the valve guides and in the slopes of the intake and exhaust faces. Cylinder heads are interchangeable between the right and left banks. Great care is taken in locating the cylinder head holding screws, so that their bosses in the block will have no contact with the cylinder sleeves. The purpose of this precaution is, of course, to avoid distortion of the cylinder bores when the head screws are tightened.

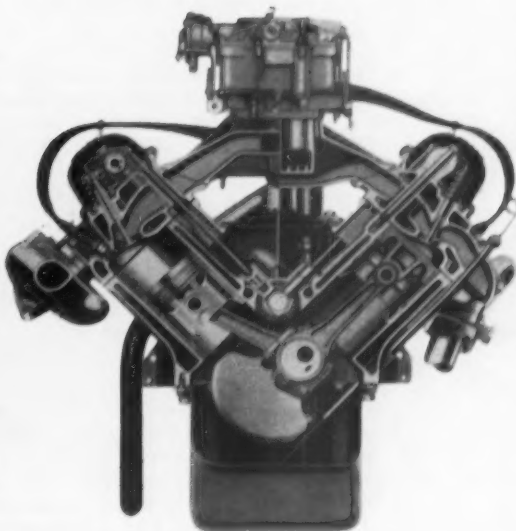
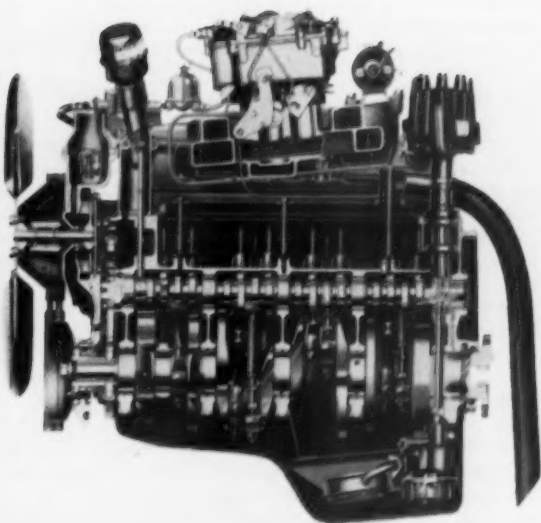
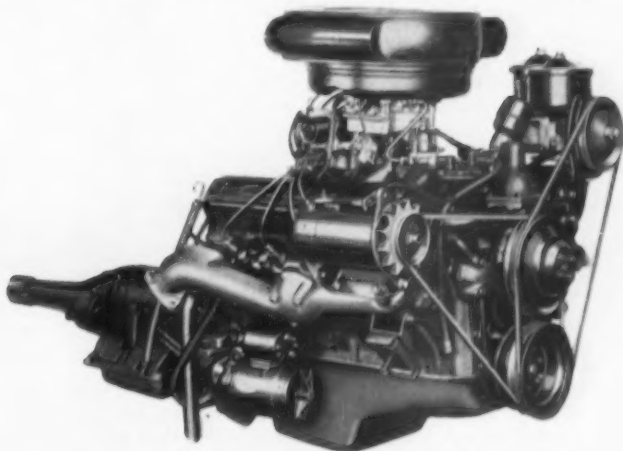
The valves are generally placed side by side more or less in the plane of the cylinder axes. This is undoubtedly the simplest and cheapest arrangement, although the valve-ports are restricted and it is difficult to ensure proper cooling of the exhaust-valve seat. In several designs, the mixture flow past the intake valve head is restricted by the close fit of the walls of the combustion chamber.

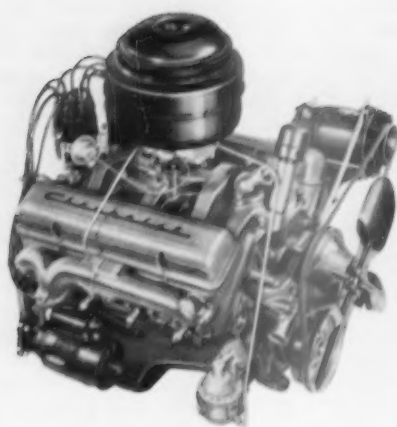
Valve gear

A neat solution to these problems is found in the Chrysler design. In these engines, the valves are arranged transversely across the cylinder and are inclined, an arrangement that has, of course, long been standard in aircraft engines. This affords room for larger valves, with excellent valve-seat stability and cooling and with excellent flow conditions about the valve heads. The sparking plug may be placed within 1 in of the cylinder axis and no part of the combustion chamber overlaps the cylinder bore, so flame travel conditions are excellent; in fact, the combustion tends to be somewhat rough. Against this, two valve rocker-arm assemblies are required for each cylinder head, there are a number of odd angles at which holes must be drilled and faced and a sealing tube is needed for each sparking plug.

An interesting compromise has been worked out for the smaller engines of the Chrysler group. In these, only one rocker-arm shaft is used in each cylinder head and the sparking plugs have been brought outside the cylinder head

The 1956 Cadillac V-8 shown in these three illustrations has 365in³ displacement





1956 Chevrolet V-8

covers. The effect of this is to reduce the cost of the engine slightly, perhaps by about £1-10-0 per unit, at some compromise to the exhaust-valve port design.

Another interesting departure from the conventional is found in the Pontiac and Chevrolet engines. In these designs, no rocker-arm shaft is used at all and each individual sheet-steel rocker arm is retained by its own stud, ball-end spacer and nut. The studs are pressed into the cylinder head and, in the original design, were drilled to carry oil from a longitudinal gallery to each rocker arm; it was later found possible to lubricate the rockers from the hydraulically-adjusted tappets. Incidentally, the rocker arms are self-aligning transversely, with respect to the valve-stem tips.

The push-rods are exceedingly simple steel tubes, with either inserted tips or rounded ends. It is noteworthy that valve-guide inserts are not used, the valve stems being directly in the head. This not only reduces cost but also lowers the temperature of the exhaust valve head by some 50 deg F, though probably at some cost in durability.

Apart from these items, the design of the valve gear is usually straightforward. Self-adjusting hydraulic tappets are used in the majority of these engines and are supplied from either one oil gallery drilled above the camshaft tunnel, or two galleries drilled alongside this tunnel. In several of the "sports" engines, the hydraulic tappets are omitted and plain mechanically adjusted tappets substituted, since the former are inclined to give trouble with pump-up at engine speeds around 5,000 r.p.m., a speed above which resonance in the valve gear and valve springs is usually troublesome.

The camshaft is driven from the front end of the crankshaft, nearly always by inverted tooth chain. A typical valve timing would be:—

I.O.	20 deg B.T.C.
I.C.	55 deg A.B.C.
E.O.	50 deg B.B.C.
E.C.	15 deg A.T.C.

Combustion chamber

The most fruitful source of disagreement concerning these engines lies in the design of the combustion chamber. All have overhead valves, almost all have compression ratios in the range between 8:1 and 10:1, but there the uniformity ends. Their design varies from a simple, "open" chamber in the Chrysler engines, to the more complicated chamber of the Buick engine, which consists of a simple enough penthouse cavity in the head, coupled with a comparatively

complex domed head on the piston, which must tend to increase the piston-head temperature.

The majority of the combustion chambers used in American V-8 engines are based on an arrangement of the valves longitudinally, in a plane parallel to that of the crankshaft and inclined at about 15 deg to the cylinder axes, so as to bring the valve stems and the overhead valve gear in toward the centre of the engine, for compactness. In the Buick design, this angle is 45 deg, so the valves are vertical when assembled in the engine. The exceptional Chrysler design has already been mentioned.

The conventional valve arrangement does not leave a great deal of latitude in the design of the combustion chamber. It becomes a problem of how best to ease the flow conditions about the intake valve head, how to place the sparking plug where it will be accessible and yet obtain a minimum flame travel, and how much of the area above the piston will be flattened off, to serve as quench area. This quench area will usually be about 45 per cent and should be kept high, since roughness is not a problem with engines of this type, well mounted in rubber supports.

Compression ratios are high, even as high at 10:1, as mentioned above. This leads to the question of machining the entire combustion chamber. With careful work in the foundry, the machining of the cylinder head located from the combustion-chamber roof and close tolerances on the crank throw, connecting rod length and so on, it is possible to control the compression ratio of an as-cast combustion chamber to about ± 0.35 of a ratio. This does not seem close enough and it is now possible completely to machine even quite complex chambers (for example the Packard chamber) without adding greatly to the cost.

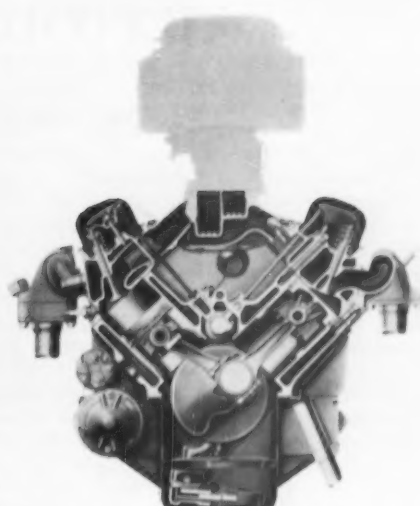
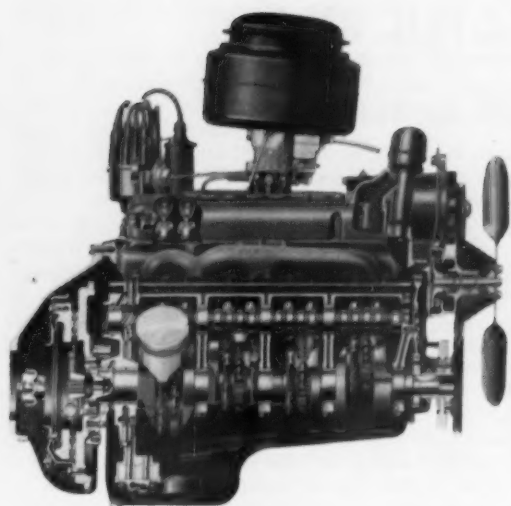
Fuel system

The only essential difference in the fuel systems of these engines is the number of carburettors fitted. These consist of either a single duplex (two-venturi) carburetter, a single quadruplex (four-venturi) or "four-barrel" carburetter, or two quadruplex carburetters. In the constant search for greater power output, the tendency is to provide the latter pair. In all cases, fuel is supplied by a mechanically-driven fuel pump and air is supplied through a large air cleaner, which has been much flattened, to permit the lowest possible body lines. In a few cases, the air is drawn from outside the engine compartment.

Intake manifolds conform strictly to one type. Numbers 1 and 4 left cylinders and numbers 2 and 3 right cylinders are supplied from one venturi (or pair of venturis) and the remaining cylinders from the other venturi (or pair). In plan view the manifold thus resembles two superimposed H's.

In the past, the two exhaust manifolds have usually been connected to a single exhaust pipe, but the tendency is to provide two separate exhaust systems, one for each bank of cylinders. Heat for the intake manifold hot-spot is invariably obtained by a short cross-connection between numbers 2 and 3 exhaust ports of one bank to the directly opposite exhaust ports of the other bank of cylinders. This connection passes through the intake manifold on its under side. Until the engine warms up, exhaust gases are forced through this connecting pipe, by a thermostatically-controlled blocking valve in one exhaust pipe. So long as this valve is closed, the exhaust from one bank must find an exit through the centre exhaust ports and the exhaust pipe of the opposite bank of cylinders.

In every case, a combined water pump and fan unit is mounted at the front end of the engine directly above the crankshaft and is driven at, or slightly less than engine speed by means of a belt. The dynamo is mounted high up, either on the right or left side of the engine and is usually



Displacement of this 1956 Chevrolet engine is 265in³

driven by the same belt. It is arranged to swing, for belt tension adjustment.

Basically, then, the accessory layout is extremely simple. Unfortunately it does not end there; there are now two other accessories which have nothing whatever to do with the engine, for which space must be found. The first is the hydraulic pump for the servo-assisted, or "power" steering gear, now fitted to about 50 per cent of all American cars manufactured. This is usually mounted on the rear end of the dynamo, so that no additional drive need be provided. Finally a compressor for the air-conditioning system must be fitted to about 10 per cent of current-production engines. This unit is mounted high up on the side of the engine opposite to the dynamo and is usually driven by a separate belt.

In America, the trend toward the use of V-8 engines is now almost universal in motor cars and is rapidly spreading into commercial vehicles. It is the result of the availability of fuels of up to 100-octane anti-knock value and a demand

for quietness and smoothness, coupled with high maximum power output, in engines of even the lowest-priced models. Automatic transfer-type machinery has made it possible to manufacture V-8 engines at a very low cost—extremely low, when measured in man-hours, rather than wages. V-8 engines are fitted to motor cars which, to American standards, provide little more than bare transportation and to cars in the very highest price class.

As to the next step, it seems likely that the V-8 engine will occupy its leading position in America for some years to come. Fuel injection will probably replace the carburettor in at least one American engine late this year and some work is being done on turbo-blowers for supercharging. Perhaps the next important move will be to gas turbines, say in about 6 or 8 years time, a development which goes steadily forward, but is held up at the moment by the difficulty of designing a satisfactory heat exchanger. Meanwhile, the V-8 engine is still undergoing intensive development and will prove difficult to oust from its present position.

1956 AMERICAN V-8 ENGINES

Group	Make	Bore and Stroke in	Displacement in ³	Rated output h.p.	Compression ratio, : 1
American Motors	Hudson and Nash ..	4.00 × 3.50	352	220	9.55
Chrysler	Plymouth	3.63 × 3.25	270	180 and 189	8.0
		3.75 × 3.13	277	187	8.0
		3.81 × 3.32	303	240	9.25
	Dodge	3.62 × 3.80	315	218 and 230	8.0
	De Soto	3.72 × 3.80	330	230 and 255	8.5
	Chrysler	3.81 × 3.62	331	225 and 250	8.5
		3.94 × 3.62	354	280 and 340	9.0
Ford Motor	Ford	3.62 × 3.30	272	173 and 176	8.0 or 8.4
		3.75 × 3.30	292	200 and 202	8.0 or 8.4
	Mercury, Thunderbird	3.80 × 3.44	312	210 to 225	8.0 to 9.4
	Lincoln, Continental ..	4.00 × 3.66	368	285	9.0
General Motors	Chevrolet	3.75 × 3.00	265	162 to 225	8.0 or 9.25
	Pontiac	3.94 × 3.25	317	205 and 227	7.9 or 8.9
	Buick	4.00 × 3.20	322	220 and 255	7.6 to 9.5
	Oldsmobile	3.87 × 3.44	324	230 and 240	9.25
	Cadillac	4.00 × 3.62	365	285 and 305	9.75
Studebaker-Packard	Studebaker	3.56 × 3.25	259	170 and 185	7.8 or 8.3
		3.56 × 3.62	289	195 and 210	7.8 or 8.3
	Packard	4.00 × 3.50	352	240 and 275	9.5
		4.12 × 3.50	374	290 and 310	10.0

AUTOMOBILE BRAKES

Considerations Affecting Fade and Stability*

J. W. KINCHIN †

BRAKING problems during the post war years have become greater as the years pass, for the reasons that vehicle weights have increased, maximum speeds have increased, acceleration has increased and to make things worse there is a tendency to reduce wheel sizes. This reduction in wheel size involves the adoption of a larger tyre section, and this in conjunction with the lower silhouettes and wrap-round mud wings has meant that the air flow over the brake drums has also been reduced. Therefore, many conditions have combined to render the provision of adequate brakes more difficult. At one period it is probable that sufficiently light pedal efforts were a problem, particularly in view of the increase in vehicle weights but on the American market particularly this has been met by the provision of vacuum boosters on many cars, while on the European market the bulk of the production consists of lighter vehicles and acceptably light pedal efforts are obtained without the need for boosters. Thus it can be stated that the pedal effort is not a problem in braking today for normal brake usage.

From the foregoing, it can be seen that the braking problem today must be mainly the ability of the brake to deal with high energy absorptions, and while some improvement to braking has been made by the provision of wider brakes giving larger lining areas, it is probably true that no major contribution to braking has been made on American vehicles since the war. This is not completely the case with European produced vehicles where braking conditions and requirements are generally much more arduous than those in America. It is generally the custom to provide a larger area of lining per lb. of gross vehicle weight, than is the case with American vehicles and the test specifications which most European vehicles have to meet are in many cases more arduous than those required of the American brake. This is because it is common to take a fade test in the U.S.A. as 12 stops from 60 m.p.h. at 0.4 mile intervals and $\frac{1}{2}g$ deceleration, while we take 12 stops from 70 m.p.h. or even 80 m.p.h. at 1 mile intervals and $\frac{1}{2}g$ deceleration.

There is one fundamental difference between the approach to braking in Europe and in the United States; this is that generally vehicles produced in Europe do not employ brakes with a high self-energising factor. The basic brake specification for Europe consists of either a two leading shoe front brake and a leading and trailing shoe rear brake, or of a leading and trailing shoe front and rear brake. Most American vehicles use a self-energising type of brake, both front and rear in which the secondary shoe is energised by the primary shoe. The typical brake factors of these different brakes are as follows:

Drum drag	
Where brake factor = $\frac{\text{Drum drag}}{\text{Shoe Tip Effort}} = C$	
(a) Two Leading shoe	$C = 2$
(b) Leading and Trailing Shoe	$C = 1.15$
(c) Self-energising Brake	$C = 5$

The figure quoted for the self-energising type of brake as used in America is a typical one and can be modified by using different grades of lining on the primary and secondary shoes with the objects of either equalizing lining wear or

increasing fade resistance by putting a lower co-efficient, high fade resistance lining on the secondary shoe. It should also be stated that if the two leading shoe and servo brakes are compared for torque assuming the same shoe centre lift for both brakes and the same pedal travel and pedal effort, then the ratio of torque is 2 for the two leading shoe to 3 for the servo type brake (approximately).

However, although by suitable design and choice of linings it is possible to reduce the servo effect and thus the power of the self-energising type of brake, it still remains that the change in torque for a given change in co-efficient of friction will always be greater on the self-energising type of brake than on the two leading shoe type, since this is inherent in the design of the brake. If it be admitted that the inability to dissipate energy at a high rate, which is generally termed brake fade, is the main problem in braking today, then it seems to the writer that the basic approach to this problem should be to adopt a brake which has fundamentally the minimum change of torque for a given change in co-efficient of friction.

Brake stability

It is interesting and also strange that whereas no British Manufacturer to the writer's knowledge has held up American braking of the automobile as a standard to be desired, it is general amongst British Manufacturers and users all over the world to hold up American truck braking as a high standard. This is significant, because a large number of American trucks use either fixed cam or two leading shoe brakes and the self-energising type of brake is not in wide use on heavy trucks. This seems to prove that if you want consistent braking and absence of fade, then a low self-energising factor should be adopted.

It is the writer's opinion that providing the pedal effort is sufficiently light then the main requirement which the brake has to meet is consistency. It must not pull to the left or the right; it must not become "grabby" or "touchy" and the pedal effort for a given deceleration must be consistent. These conditions are of course quite easy to meet if a lining is available with a co-efficient of friction which is absolutely consistent against temperature. Unfortunately no such brake linings exist at an economic price.

It is also the writer's opinion that a slight fall off in co-efficient of friction against rise in temperature is acceptable, and indeed desirable, but a rise in co-efficient with a rise in temperature is to be avoided. The reason that a fall in co-efficient of friction is stated to be desirable, is because this can give some indication that the brakes are being heavily used and since the pedal effort would become heavier it would tend to restrict the use of the brakes.

Shoe geometry in relation to lining co-efficient

The reason that a rise in co-efficient of friction is detrimental, is because the brakes tend to become erratic and "grabby" and also as is well known, in designing a brake it is desirable to put the lining in the best position for power and maximum utilization of the available volume of the lining. This is in order to give the maximum lining life and to spread the load more evenly over the lining area, so as to avoid concentrated peak loadings and reduce fade. Also if sliding

* From a paper presented before the SAE at Detroit.
† Director and Chief Engineer, Girling Limited.

shoes are employed which can centralize themselves in the brake drum the optimum position for the lining has to be determined with reference to the co-efficient of friction and the wear pattern is altered with maximum wear moving towards the heel or toe of the shoe as the co-efficient of friction either rises or falls.

If the lining position for a given co-efficient of friction is chosen to give a reasonable compromise on power, taking into account the maximum co-efficient of friction at which self-locking would occur, then it is desirable that this co-efficient of friction should not be exceeded and if allowance for a large percentage increase in co-efficient of friction has to be made to cover the brake when it is hot, then the performance at normal temperatures has to be sacrificed. With regard to brakes which become erratic when hot, it is interesting to observe that the locking angle of the leading edge of the lining of many brakes closely approaches the friction angle of the lining and study of the trailing end of the lining shows that the locking angle is often within this figure. This means that in the forward direction a moderate rise in co-efficient of friction would give a locking or grabbing brake and in reverse the tendency would be worse. In order to avoid this tendency it is necessary to avoid long lining arcs in most brakes, or if long arcs are used to employ low co-efficient linings of good stability. We have found it to be good practice to arrange the length of the lining and position such that the tangent of the locking angle of the leading and trailing ends of the lining is greater than the co-efficient of friction of the lining and this infers arcs of 90-100 deg.

Two trailing shoe brakes

Admittedly if brakes with a low self-energising factor are adopted the external power to the brakes has to be increased and prior to the introduction of vacuum boosters it would not have been possible to brake cars at the rates existing today with acceptably light pedal pressures, purely with the driver's effort and taking into account female drivers.

However, the introduction of vacuum boosters has solved this problem, and it is considered feasible today to use a brake of lower self-energising factor with a larger booster, thus to obtain extremely consistent results. The pedal efforts of

course can be made whatever is desired by choosing the correct area of booster cylinder.

In essence the two trailing shoe brake is a two leading shoe brake run in reverse. There is, therefore, an anti-servo effect and the brake factor is reduced. In detail it consists of two shoes each operated by a hydraulic cylinder and the shoes are conformable by having the web depths reduced to a certain pattern. The brake is self-adjusting on the hydrostatic principle, the hydraulic pistons following out the shoes as the lining wears and the volume being made up by hydraulic fluid from the reservoir, there being no trapped line pressure. Frictional anti-shake back devices are used on each shoe to restrain them from vibrating away from the drum.

Brake size and horsepower absorptions

As mentioned above it is the tendency in Europe to use larger brake areas for a given vehicle weight, than is the case in the United States and the figures in the accompanying Table make a comparison between common practice on European vehicles and practice in America today:

These figures show:

Considering front brakes only

(a) Although the mean horsepower absorption figures for both American and British cars are approximately equal, the horsepower absorption figures for the secondary shoe on American brakes is approximately 3.42 average while that of the British vehicles is 2.45 average per shoe.

(b) The horsepower absorption per in² of drum area is lower on the British cars.

(c) On the faster and heavier British cars two trailing shoe brakes are used and horsepower figures are higher as short lining arcs are used.

Considering rear brakes only

(a) American cars tend to use smaller rear than front brakes in proportion to the weight and braking distribution.

(b) The secondary shoe horsepower absorption per in² of lining is on average considerably above that of the leading shoe of the British vehicle (3.33 to 2.355).

(c) The horsepower absorption per in² of swept drum area is approximately 40/50 per cent greater on American vehicles.

COMPARATIVE H.P. ABSORPTIONS

Vehicle	G.V.W. lb	Braking Ratio F/R	H.P. Absorption in ² Lining						Speed M.P.H.	H.P. Absorption per in ² Drum Swept Area		
			Front			Rear				Front	Rear	
			Mean	P.S.	S.S.	Mean	P.S.	S.S.				
A	4342	55.8/44.2	2.43	1.39	3.24	2.46	1.4	3.28	90	1.48	1.5	AMERICAN
B	3830	55.8/44.2	2.66	1.5	3.58	2.42	1.36	3.26	90	1.62	1.46	
C	3871	62.2/37.8	2.32	1.16	3.26	1.84	0.97	2.77	90	1.61	1.26	
D	5235	53/47	2.72	1.56	3.62	3.01	1.72	4.0	90	1.67	1.87	
A	3600	66.8/33.2	2.3			2.3 L/s			90	1.21	0.8	BRITISH
B	3388	65/35	2.42			1.95 L/s			90	1.48	0.8	
C	4100	58/42	2.83			2.69 L/s			90	1.47	1.06	
D	4462	58/42	3.02			3.12 L/s			90	1.58	1.15	
E	3059	65/35	1.88			2.16 L/s			70	1.14	0.88	
F	2800	65/35	2.24			1.92 L/s			65	1.39	0.75	

AMERICAN

BRITISH

In calculating H.P. figures on duo-servo brakes it has been assumed that work done by primary shoe to secondary shoe is in the ratio of 1 to 3. All figures based on 28 ft/sec/sec deceleration.

(d) It is possible satisfactorily to use higher horsepower figures for the leading shoe, providing the horsepower dissipation figures for the drum are kept round 1-1.2 per in².

The question, therefore, arises as to what is the correct area of brake to be used for a given duty. It is the practice of my Company to assess the mean horsepower loading per sq in of lining from a vehicle speed slightly below the maximum given speed of the vehicle and for a deceleration of 28 ft sec². We normally take a speed of 0.9 approximately of the maximum speed of the vehicle and this is chosen bearing in mind the price range of the brake and the deceleration to be accepted, and we attempt to limit the mean horsepower absorption to a maximum of 2.5 per in². It is not considered practical to take decelerations from the maximum speed of the vehicle, since taking into account wind resistance and time lag the brakes cannot be applied at the maximum speed. In the case of the two leading shoe or two trailing shoe brakes, this absorption is the same for both shoes but in the case of the rear brake where leading and trailing shoes are employed we consider the horsepower absorption on the leading shoe only.

Experience has shown that with the quality of linings which are available today, if 2.5 horsepower per in² of lining is not exceeded, fade is not usually a problem with the normal amount of cooling which is present. If, however, fade should be a problem with two leading shoe brakes then two trailing shoe brakes are employed. We normally use 110 deg lining arc, but recent work on high speed cars has shown that an 85 deg lining arc gives improved fade resistance and although the horsepower absorption per sq in of lining has increased in the ratio to the decrease in lining arc, stability has been improved.

The characteristic of judder or high speed vibration which is sometimes experienced when brakes are applied from high speeds with a high deceleration has also been considerably improved, by reaching the lining arc, and it has been found that by making the shoe web conformable in a certain pattern which has to be carefully determined, the high speed vibration is eliminated or very much reduced. It appears that making the shoe web conformable to this pattern permits the lining to spread the work equally over its whole area, and this eliminates blue spotting. Advantage is also obtained from the fact that a larger arc of the drum is left uncovered with the shorter linings and permits better heat dissipation. We have also found that the decrease in lining arc and, therefore, volume has not materially reduced the lining life under these conditions.

Disc brakes

Much interest is being displayed at the present in disc brakes and all manufacturers are seeking a satisfactory solution to the disc brake problem. My own Company has been involved in the development of disc brakes for some time and has experienced a considerable amount of success with these brakes in racing. We also have the first passenger carrying bus in the world to go in production with these brakes; this is operating very satisfactorily and the lining life is better than that experienced with drum brakes.

It is interesting to compare the characteristics of the disc brake with those of the two trailing shoe brake. I have chosen the two trailing shoe brake because from the fade and stability point of view it is the best drum brake produced and, therefore, I think it is not worth while comparing the disc brake with any other design than this. The brake factor of the two trailing shoe brake for a co-efficient of friction of 0.38 is 0.59 and let us assume a pedal effort of 100 lb. If the co-efficient of friction falls to 0.25 the brake factor becomes 0.45, i.e. 24 per cent change and the pedal effort would rise to 131 lb; the percentage change of pedal effort

is therefore 31. Under the same conditions the two leading shoe brake factor changes from 1.75 to 0.98, that is 44 per cent and the pedal effort rises to 179 lb, i.e. 79 per cent increase. The servo brake has a still greater change. The disc brake has no servo factor but is a Girling construction using a hydraulic cylinder applying a pad on each side of the disc.

The brake factor is equivalent to the co-efficient of friction which is therefore 0.38 in the case of a co-efficient of friction of 0.38 and becomes 0.25 if the co-efficient of friction drops to 0.25, and the pedal effort changes from 100 lb to 152 lb. This is a percentage change in the brake factor of 34.4 per cent and 52 per cent increase in the pedal effort. Therefore, purely on theoretical grounds the two trailing shoe brake is less fade conscious than the disc brake; this is not substantiated in practice because of the different construction of the two units.

On the two trailing shoe brake the heat has to be conducted away through the drum, while on the disc brake the heat is mainly dissipated to the atmosphere; also the uncovered area of the disc is greater than the uncovered area of the drum brake and, therefore, dissipation is better on the disc brakes. However, it would seem from the theoretical figures that a fade resistance comparable with the fade resistance of the disc brake can be achieved on the two trailing shoe design. The disc brake of course scores from the point of view of smoothness because there is no change in the shape of the disc, whereas the brake drum shape changes when being subjected to shoe pressures at high temperatures, and of course the disc brake does not expand away from the linings as is the case of the shoe brake. On the contrary the disc expands towards the linings.

Although the theoretical brake factor of the disc brake is lower than that of the two trailing shoe brake, this can be compensated for by the fact that a high overall ratio of the hydraulic system can be used with the disc brakes, owing to the fact that linings can run at very small or no clearance from the disc and the disc expands towards the linings. The Girling design uses the hydrostatic principle of automatic adjustment mentioned above.

On the drum brake the overall ratio which can be employed is limited by the ability to follow up a hot drum. Even if an automatic adjuster is used in the two trailing shoe brake it is still necessary to provide sufficient shoe centre movement in order to follow up the growth of the drum during one maximum stop, although adjustment would immediately take place when the foot is removed from the brake pedal.

Comparison of the figures for horsepower absorption for the disc and drum brakes is interesting. For the disc brake we employ a segmental lining and use a horsepower absorption figure of 6.5 to 7.5 per in² of lining. The horsepower dissipation for the disc is 0.65-0.75 per in². Thus while the lining loading has gone up by approximately three times as compared with the drum brake the dissipation rate has been almost halved per sq. in of swept area of disc compared with the drum. Although, based on these figures, it will be seen that the lining area of the disc brake has been reduced in the ratio of 3:1 the total volume of lining available for wear on the disc brake exceeds that of the comparable drum brake by some 10-15 per cent because the whole of the lining volume can be worn away, while on the drum brakes the ends of the lining arc are usually left thicker than the middle portion of the arc.

Finally I would like to sound a note of warning regarding the ability of the disc brake to rescue designers from the difficulty of the small wheel. The disc brake, in order to give satisfactory life must be chosen of adequate size and in its present state of development offers no universal panacea for designers' space problems.

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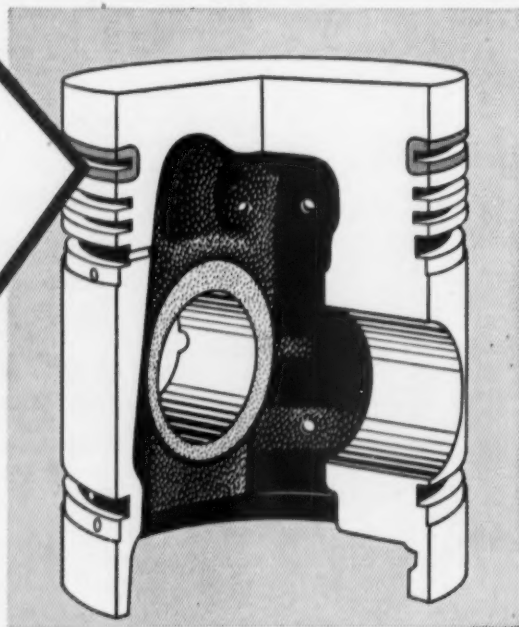
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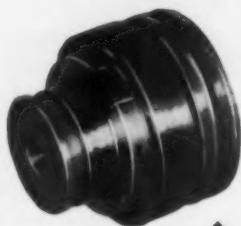
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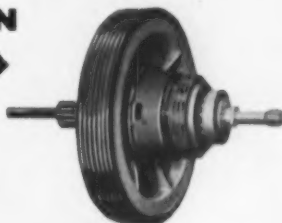


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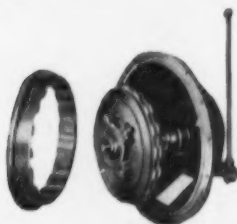
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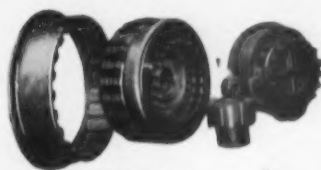
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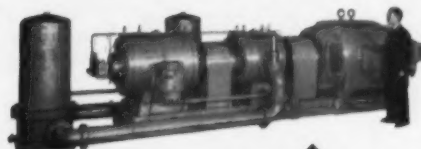
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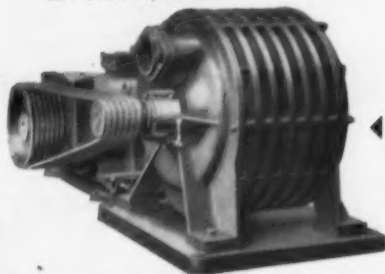
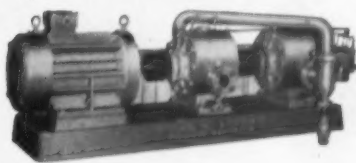


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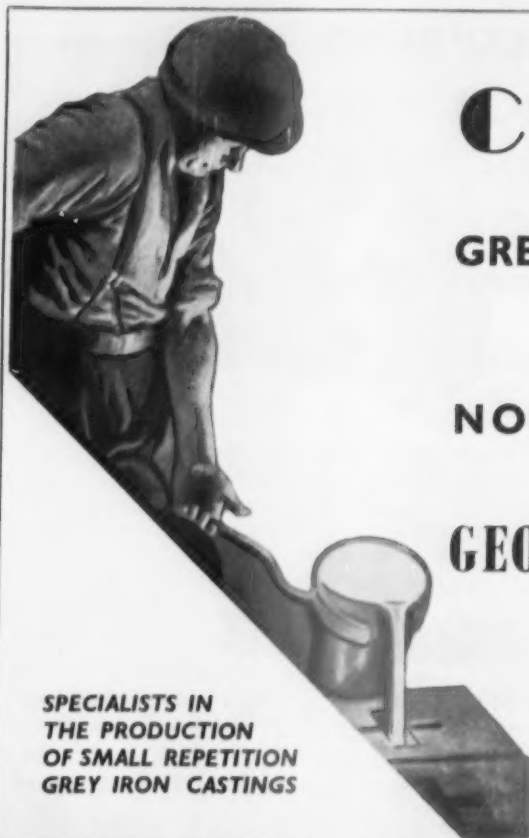
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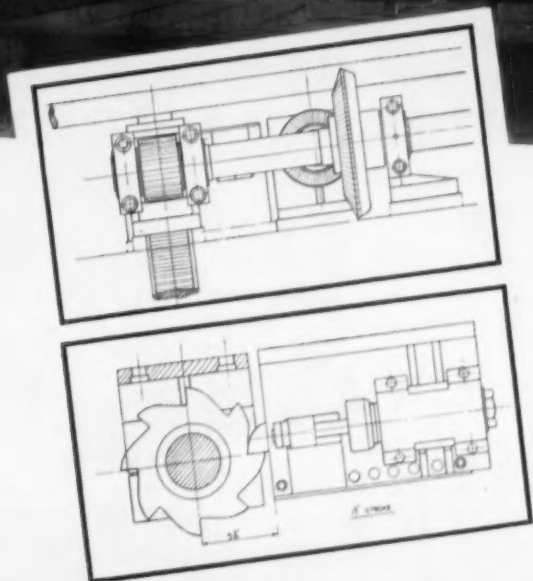
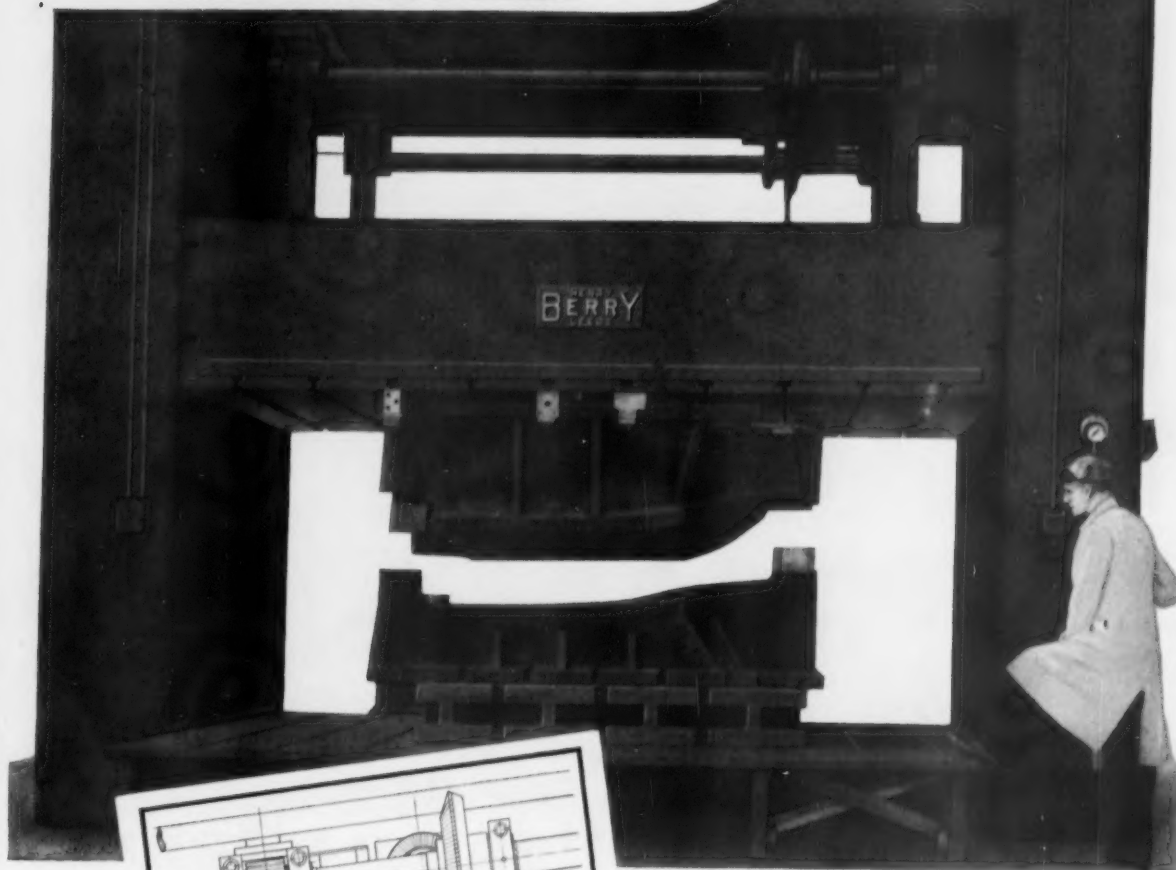
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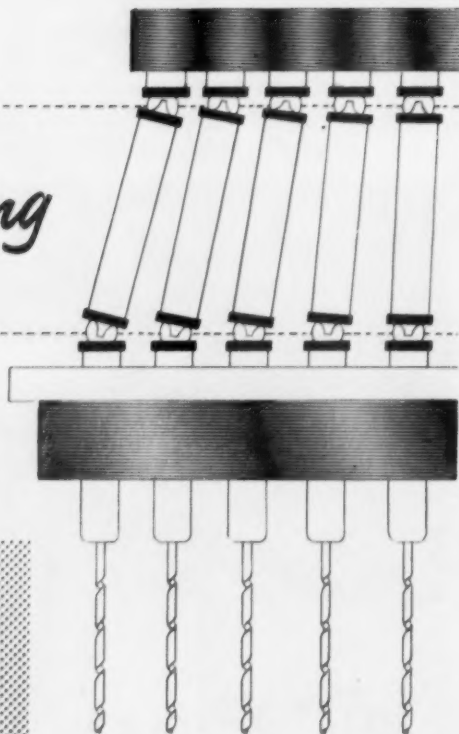
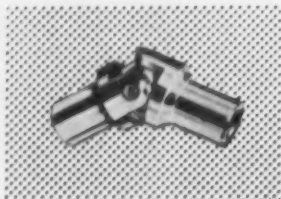
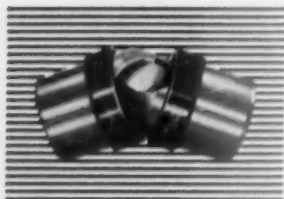
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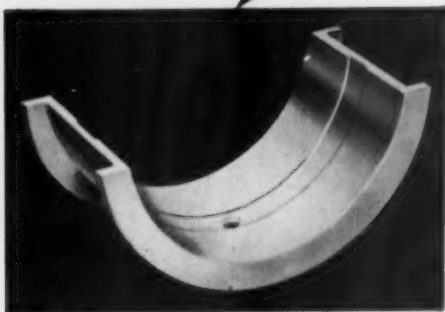
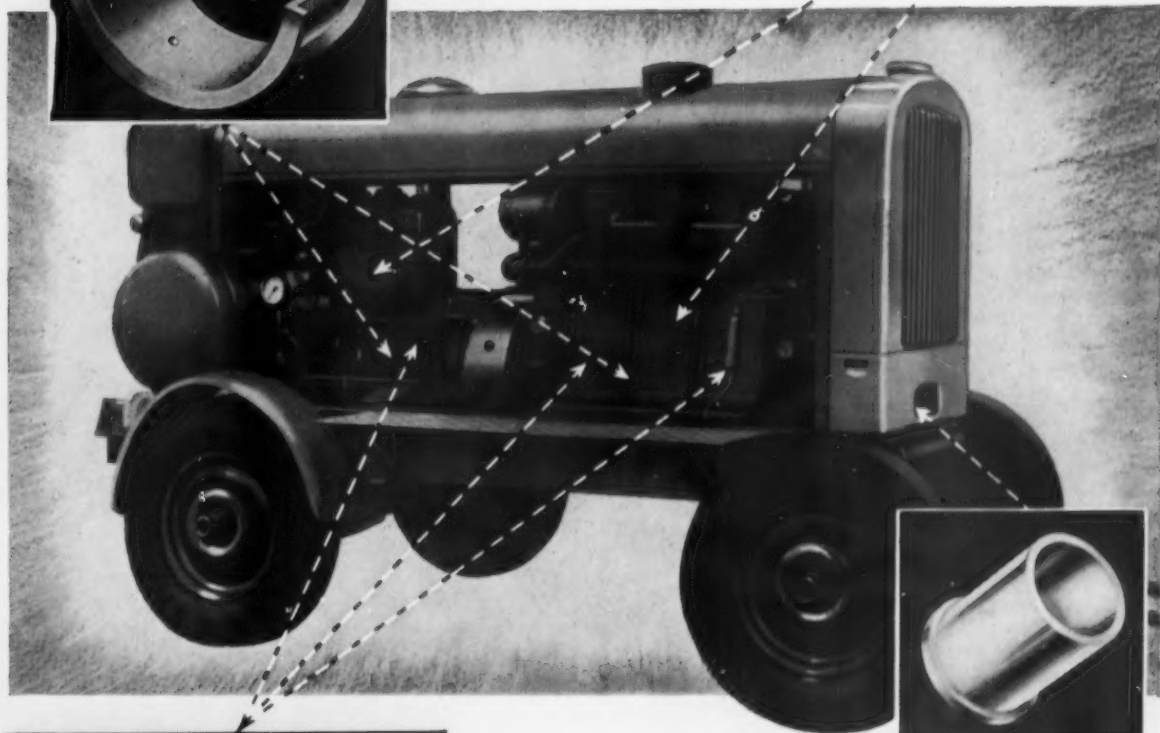
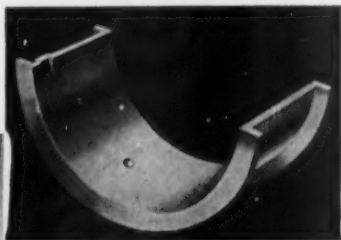
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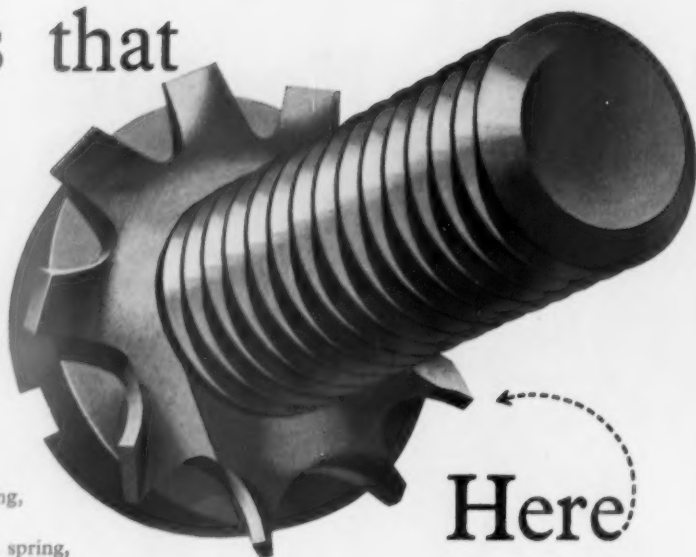
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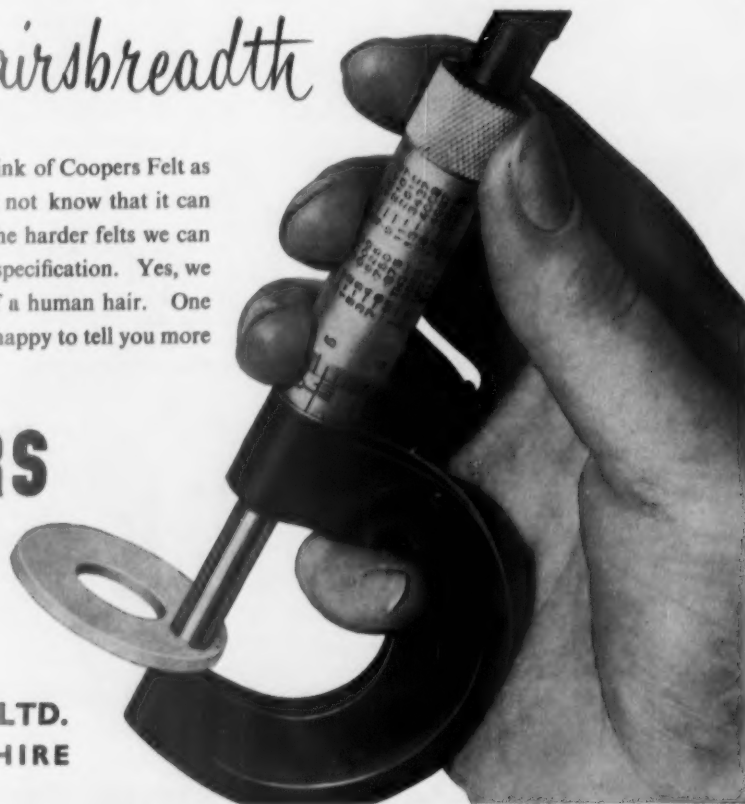
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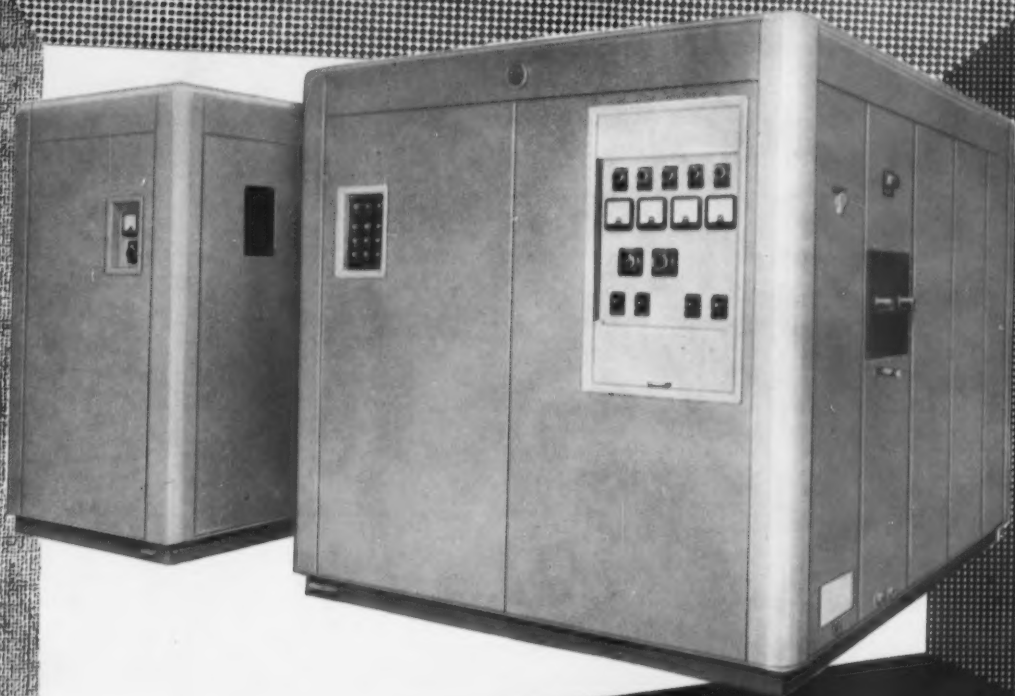
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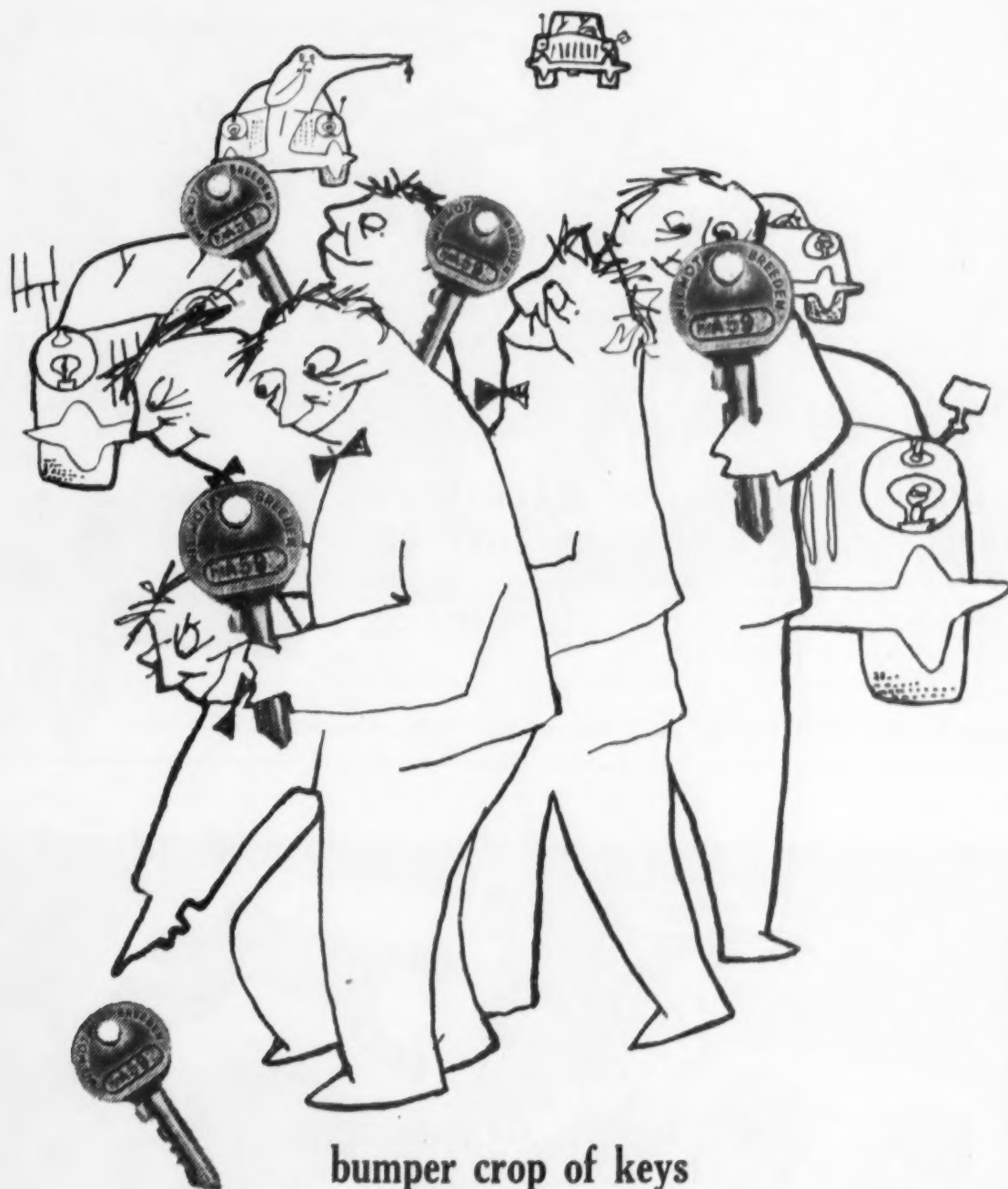
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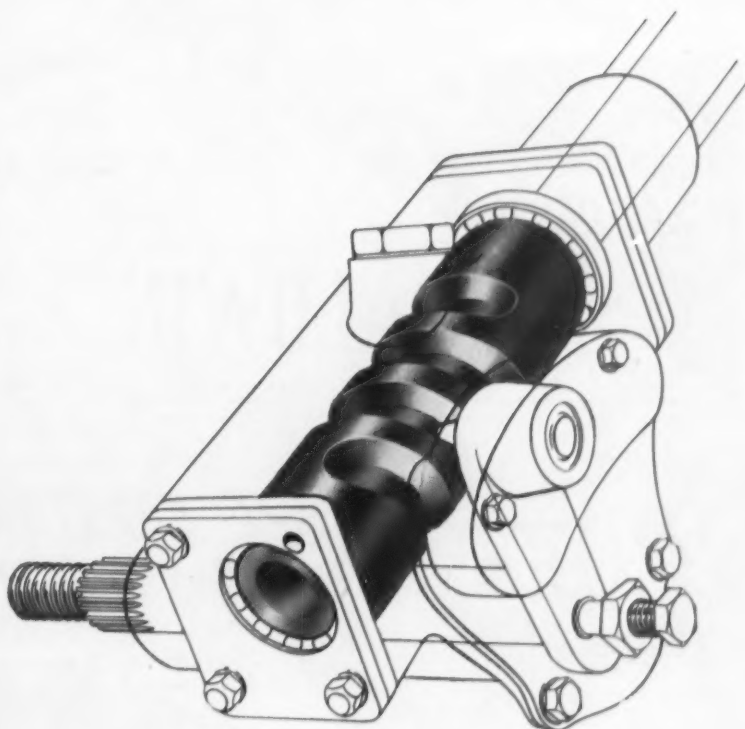
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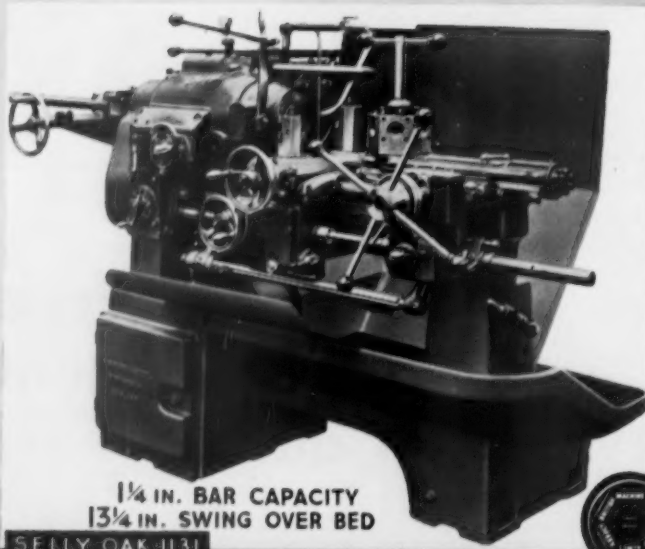
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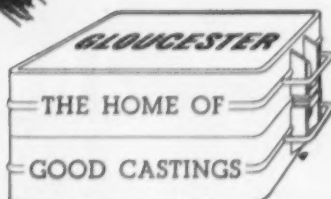
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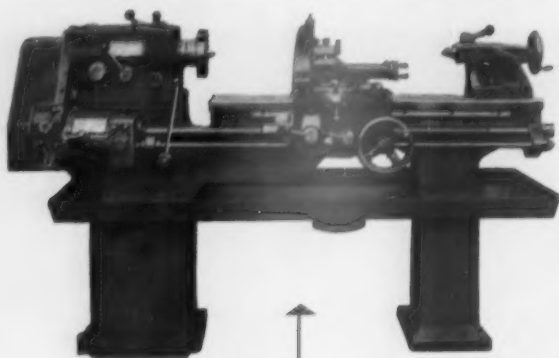
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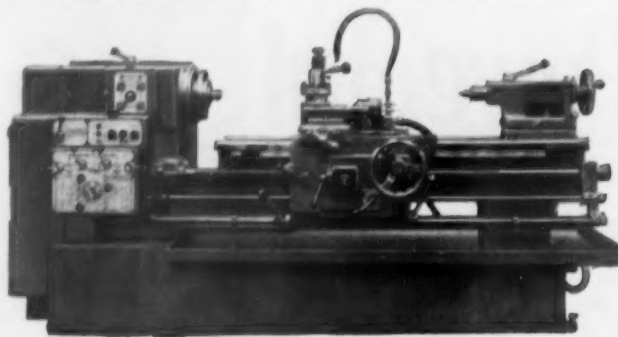
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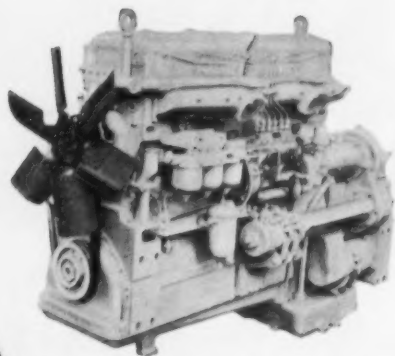
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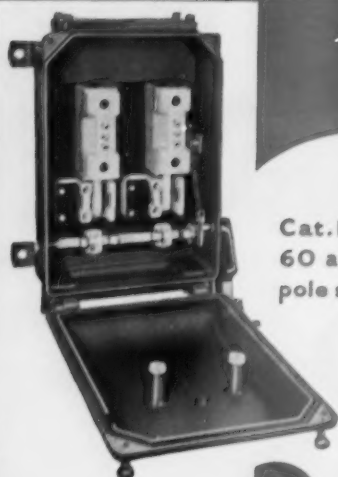
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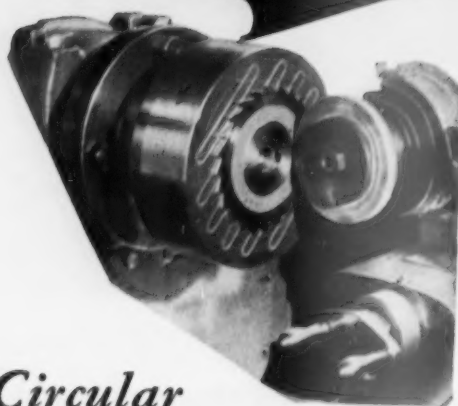
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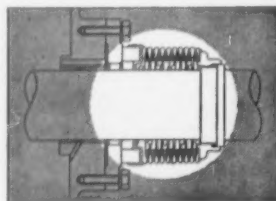
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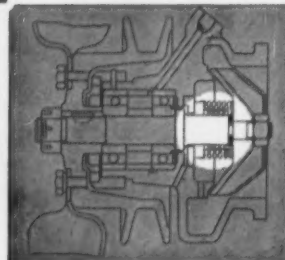


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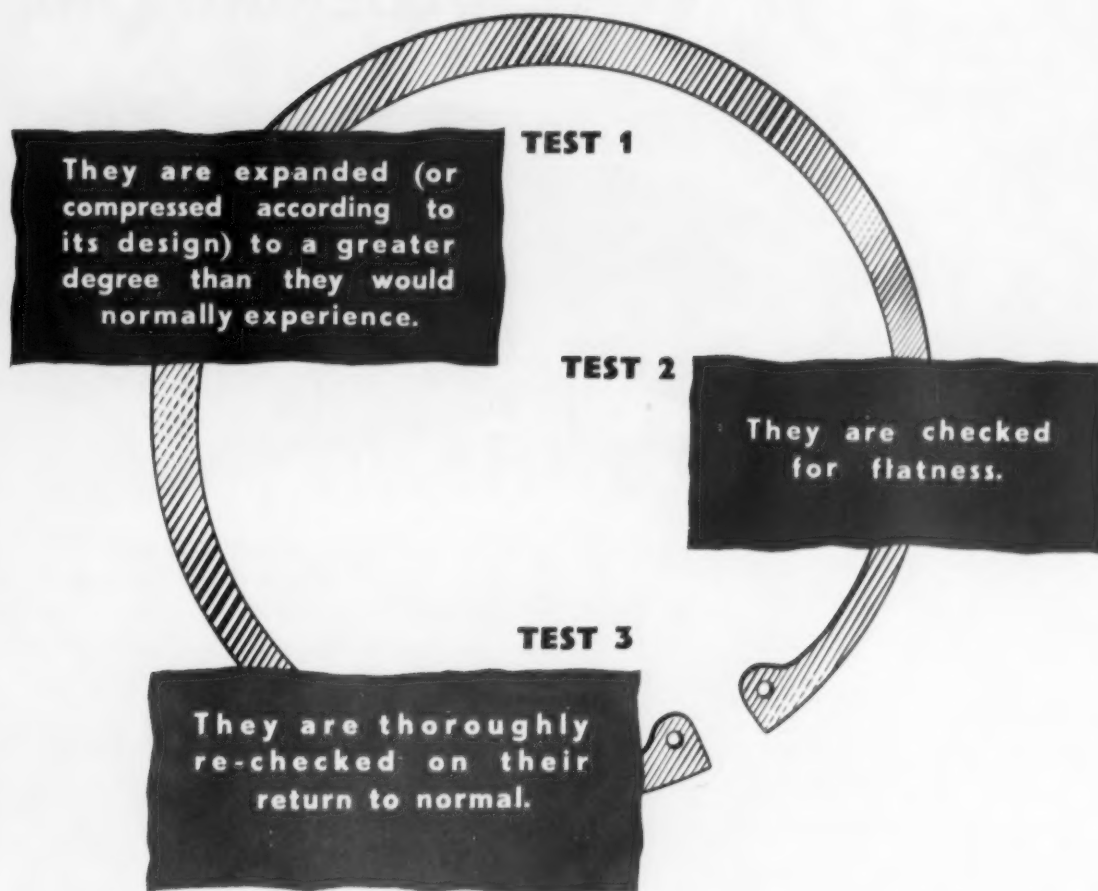


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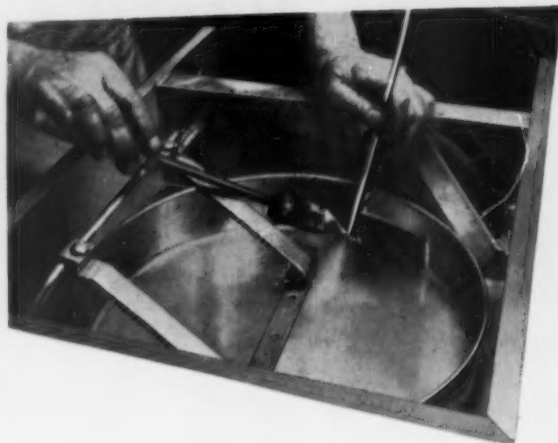
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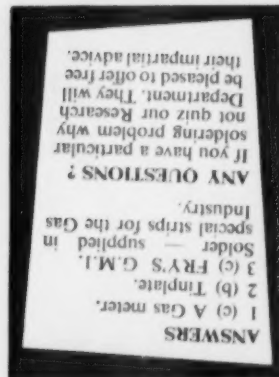
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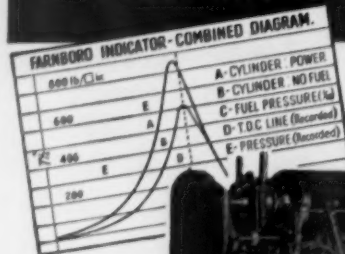
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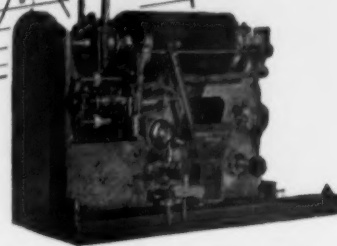


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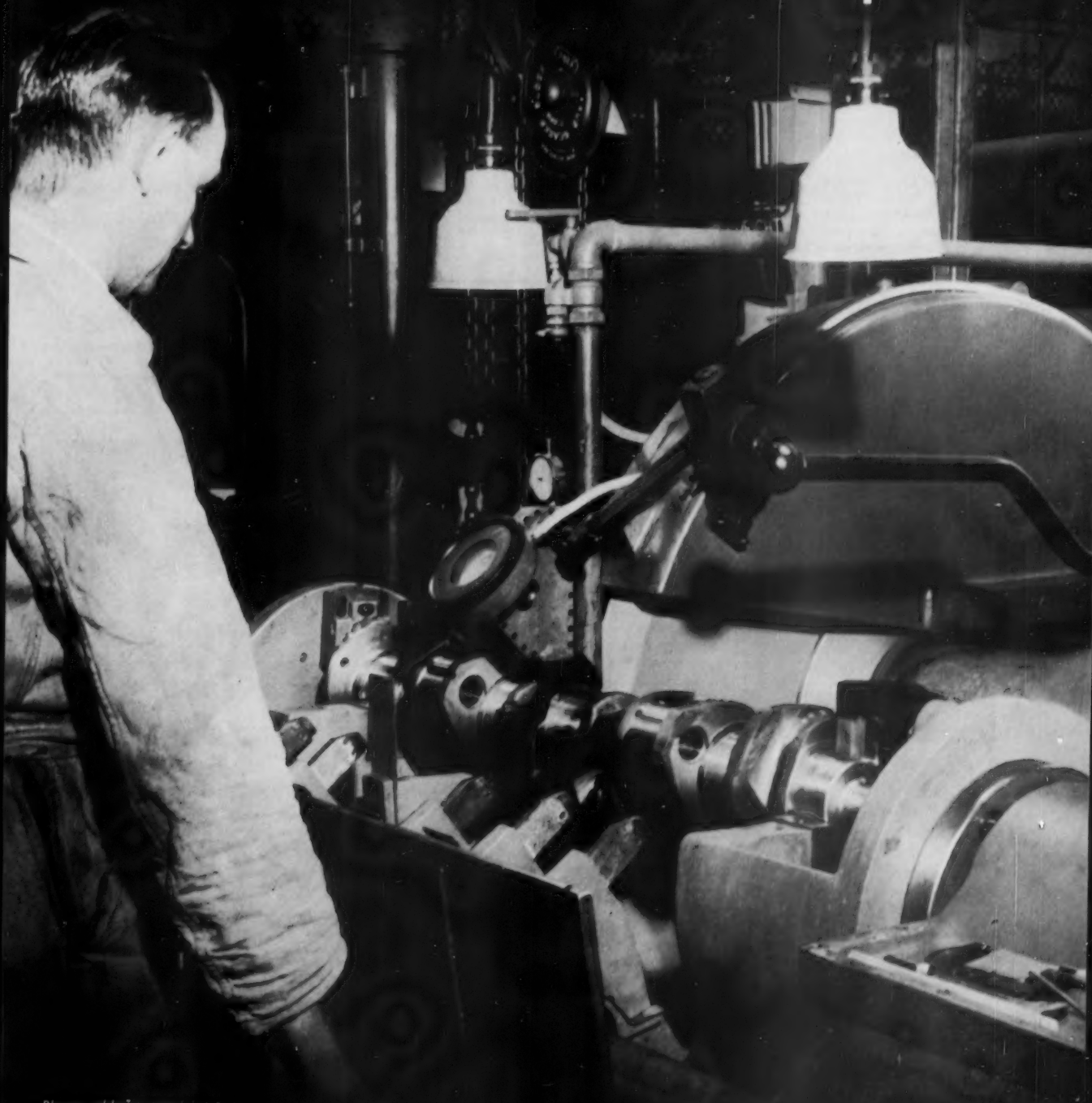
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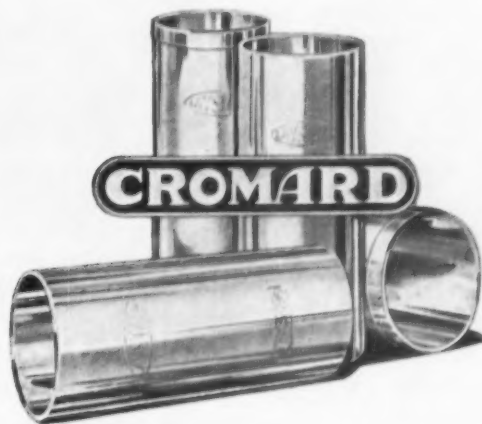


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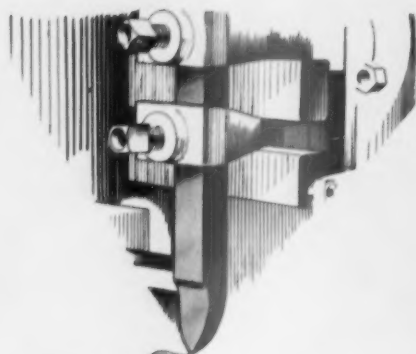
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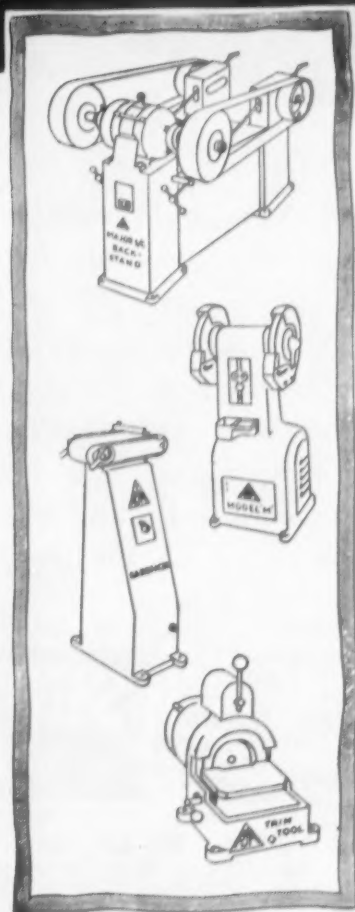
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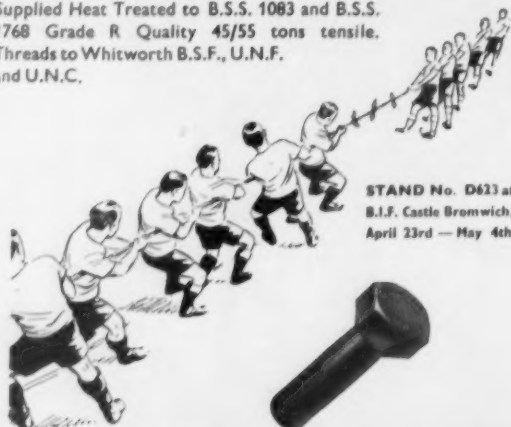
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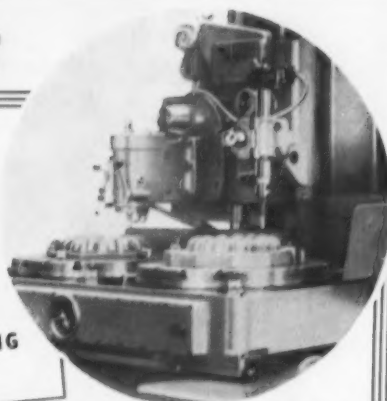


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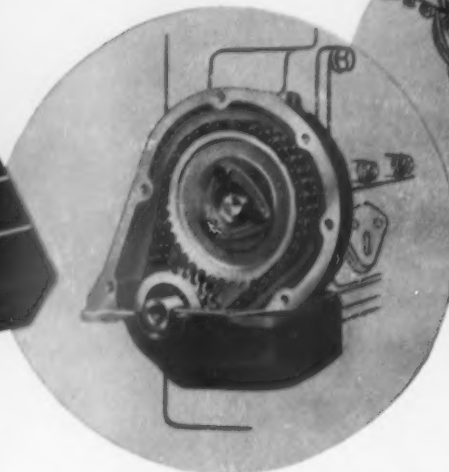
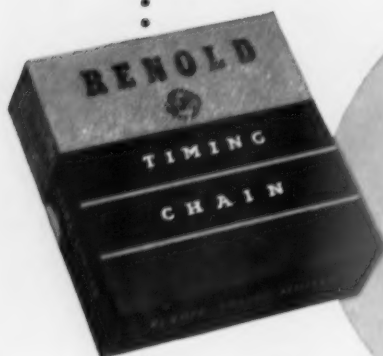
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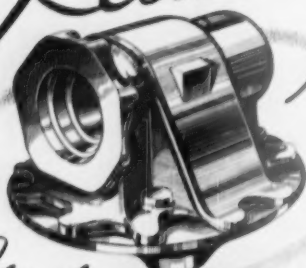
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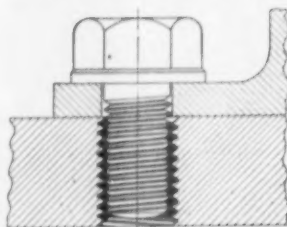
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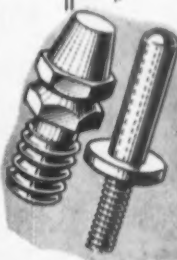
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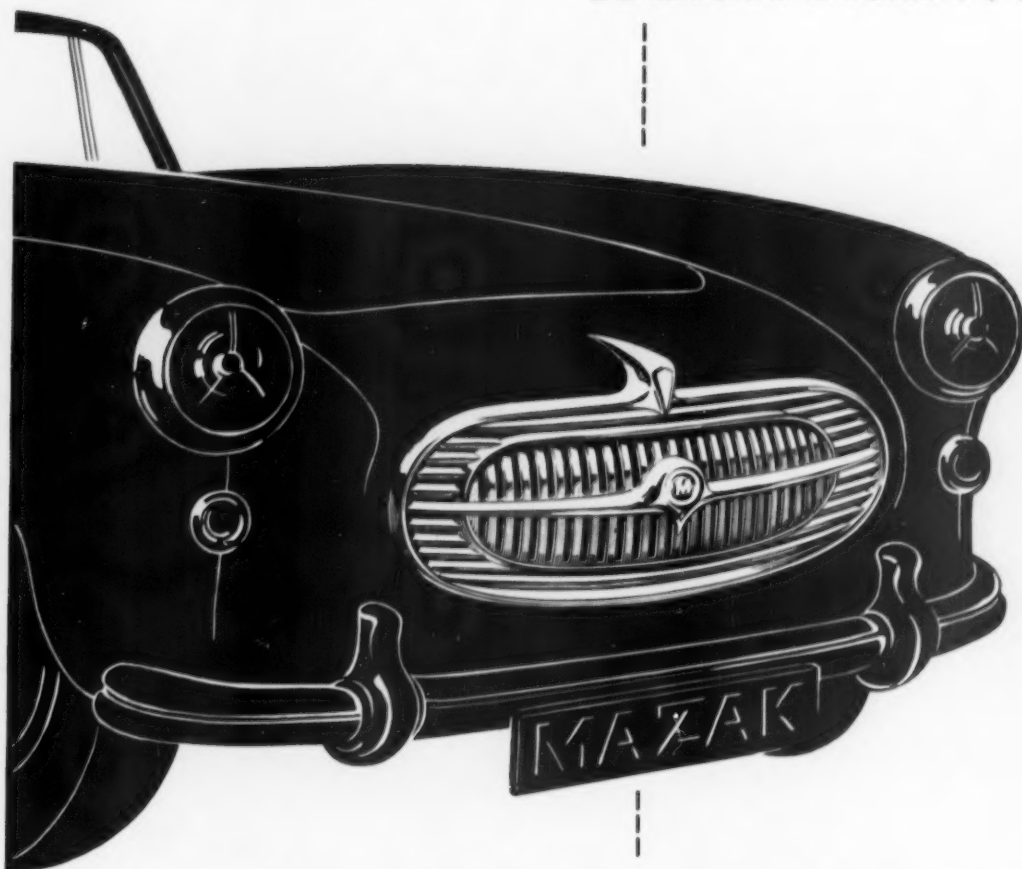
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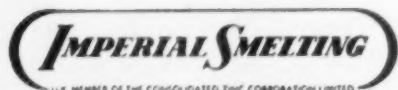
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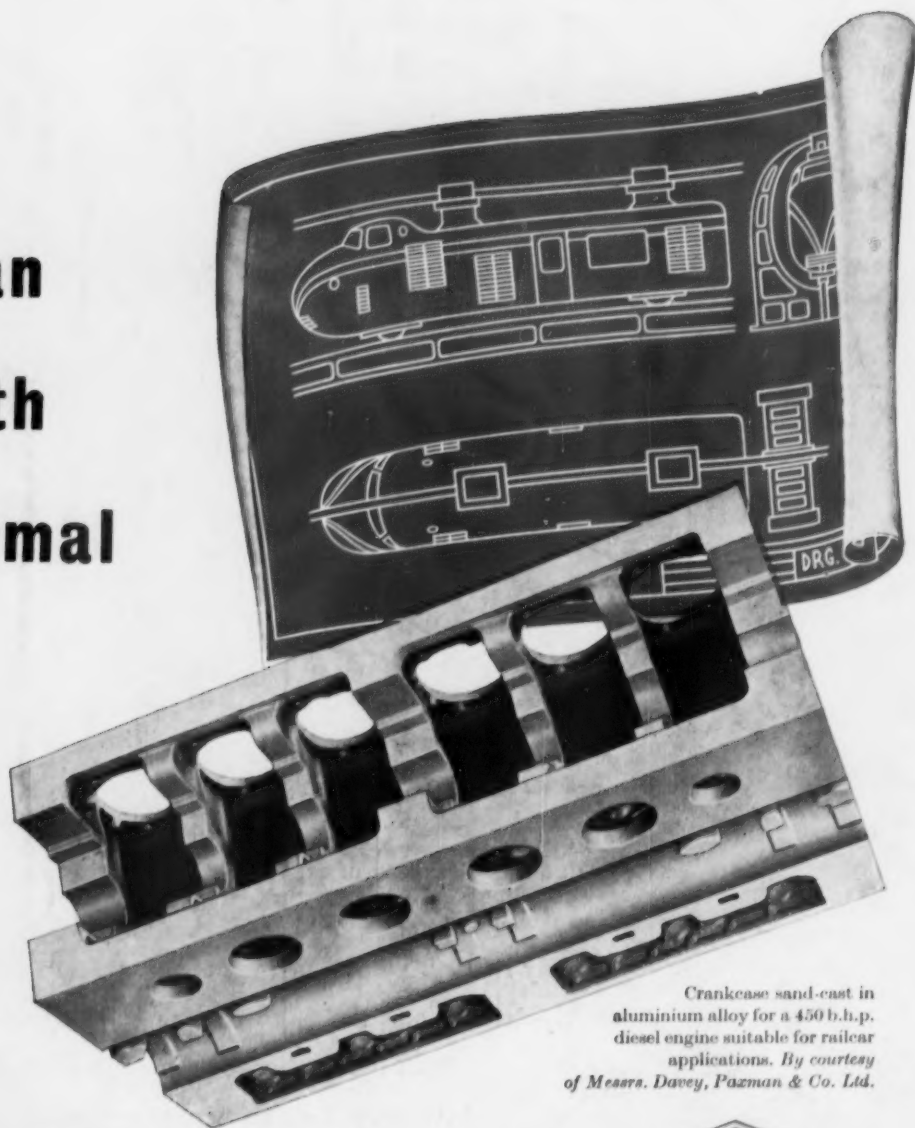
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